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# THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

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JULY 1922

SOCIETY OF AUTOMOTIVE ENGINEERS INC
29 WEST 39TH STREET NEW YORK

# A New and Wonderful Motoring Luxury

KNOWN to hardly anyone nine months ago, Stabilators have today reached a popularity throughout 28 States which is little short of a miracle —a miracle in even the Automotive Industry. The reason is simply this—

## *Stabilators are not Shock Absorbers*

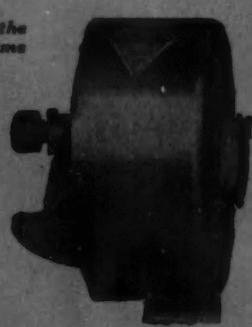
It has at last been made clear that a motor car does not need additional shock absorbers. Tires and springs properly inflated and properly lubricated give all the shock absorbing or cushioning effect which is necessary.

Heretofore however no attention has been paid to getting rid of the bump energy which the car springs absorb upon being compressed. The springs themselves hold this energy for only a moment and then recoil and pass it on up to the car body and passengers.

Stabilators are energy dissipators. By means of friction they convert a proper proportion of this spring-stored energy into heat and radiate it off. With this accomplished, the bump is passed over and the passengers are little if any disturbed.

The riding comfort and car control produced by this dissipation of energy is beyond any form of imagination. Motoring is immediately converted into a luxury which has not heretofore been dreamed of. Stabilated motoring is a form of recreation which no man will do without once he knows all it means. Cars are bought for comfortable riding. But you can never make a car comfortable until you get rid of—dissipate—the spring-stored energy which is the cause of discomfort.

*Look for the Silver Name Plate*



***Exactly opposite to snubbing***

*In checking spring recoil, Stabilators work exactly opposite to the snubbing principle. Instead of checking with a jerk at the tail end of the recoil movement, Stabilators get on the job at the very beginning of the movement and smoothly ease you back to normal. Results produced by the one method give no conception of those produced by the other. They are different to the point of absolute oppo-[illegible]*

JOHN WARREN WATSON COMPANY
Twenty-fourth and Locust Streets
Philadelphia

**WATSON**

# STABILATORS

**CONQUER ALL ROADS**

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. XI July, 1922 No. 1


# The Summer Meeting

SELDOM does one find such a universal feeling of genuine satisfaction among the attendants at a national convention as that which predominated throughout the period of the Summer Meeting of the Society at White Sulphur Springs, W. Va., June 20 to 24. The attendance, though somewhat smaller than that of the past two or three years, reached 531 and all sections of the country and branches of the industry were represented. Contrary to the expectations of many, the weather conditions were ideal. The temperature never rose above 75 deg., the days were bright and clear and the evenings unusually cool. Better climatic conditions simply could not be desired. The cuisine, facilities and hospitality at the Greenbrier Hotel were as near perfection as any of our most famous hostelries can approach. The beautiful grounds with their background of picturesque mountains formed an appropriate setting for one of the most successful Summer Meetings the Society has ever held. An unusually large percentage of the members arrived early on Tuesday and remained through the entire meeting until Saturday afternoon. Every phase of the program was received with enthusiasm. The technical papers contributed much valuable engineering information and aroused a very active discussion. The sports and entertainment programs seemed to keep all busily engaged and dull moments were banished. One heard nothing but complimentary comment on the location and the facilities on every side.

### Standards Committee Session

The regular session of the Standards Committee, which convened at 10:30 in the morning on Tuesday, June 20, was attended by 128 members and guests. After brief introductory remarks, and the declaration of a quorum by President B. B. Bachman, Standards Committee Chairman E. A. Johnston called for the reports of the Divisions as printed in the June issue of THE JOURNAL. There was also a report by the Lighting Division on the revision of the present S.A.E. Standard for Head-lamp Illumination, which will be printed in the August issue of THE JOURNAL.

The supplementary report on Motor-Truck Front-Axle Hubs was approved, with the supplementary table of ratings amended so that the former heading "Spindle load in lb. on solid-tire rating at ground" reads, "Assumptions on which calculations for spindle sizes were based."

The reports on Ignition-Distributor Mountings, Magneto Mountings and Starting-Motor Flange Mountings were approved as printed. The report on Breaker-Contacts was amended by omitting the reference to No. 8-40 thread size and approved.

The report on Tractor Drawbar Adjustments as presented was approved.

The report on Flywheel Housings was approved as printed. The report on Crankcase Drain-Plugs was discussed at considerable length. In view of the expressed opinion that this subject has to do with design rather than standardization for interchangeability, it was referred back to the Engine Division for further consideration toward its being discontinued. The report on Motorcycle Carbureter Flanges was withdrawn by Vice-Chairman R. J. Broege, due to criticisms that had been received. The Engine Division is to reconsider this subject.

The report on Leaf-Spring Steel was approved as printed. The report on Steel Spring Wire was amended by omitting the reference to "round, cold-drawn wire up to 3/16-in. diameter, except for some types of springs used in clutches, which are hot-rolled," and omitting the word "helical" from the caption to the table.

The reports on Head-Lamps, Electric Incandescent Lamps, Electric Incandescent Lamp Voltages and Motorboat Lighting Voltages were approved as printed. The supplementary report on Automobile Electric Head-Lamp Lighting Specifications was discussed at considerable length. A motion to endorse the complete report of the Illuminating Engineering Society on the Rules Governing the Approval for Headlighting Devices for Motor Vehicles, dated February 1922, with a reference added as to the acceptance by one State of tests approved in another State, and omitting the paragraph under "Approval," which refers to "Tilting Devices," was lost.

The principal reasons given against the recommendation of the Lighting Division were that portions of its report dealing with other than laboratory tests were nontechnical, intended primarily for regulatory purposes, and were outside the function of the Society. The report of the Division, including Parts I and II, was finally

approved. As this report was not published in the June issue of THE JOURNAL, it will be given in full in the August issue.

The report on Aluminum Alloys was approved as presented. The report on Wrought Non-Ferrous Alloy Specifications Nos. 77, 78 and 82 were approved as printed, except for omitting from the captions mention of the purpose for which these specifications were formulated. It was felt that the latter information should be given in sub-captions or footnotes. Specification No. 83 was referred back to the Non-Ferrous Metals Division for joint consideration with the Electrical Equipment Division, in view of the work which is in progress in the latter on standard specifications for magnet wire. The report on White Bearing Metals was amended to include certain corrections in the percentages for Specifications Nos. 10, 10A, 11, 11A, 13 and 13A. The Specifications Nos. 13 and 13A as printed were the same as Specifications 14 and 14A through error.

The report on Flywheel Pulley Lugs, as submitted by the Stationary Engine Division, was approved.

The Parts and Fitting Division's reports on Passenger-Car Front Bumpers, Rod-Ends, Plain Steel Washers, Ball-Studs, Serrated-Shaft Fittings, Tank and Radiator Caps and Lock Washers were approved as printed. The discussion on Rod-Ends developed the suggestion that the Division consider the extension of the standard to include a series of even heavier rod-ends for truck application, it being stated that the present standard sizes provide rather small pin-bearing lengths and diameters. The Screw-Threads Division's report on Screw-Threads was approved as printed. The report on Gages and Gaging, which was proposed for general information only, was referred back to the Division for further consideration in view of the criticism that it did not deal adequately with gaging for errors in lead.

The report on Top-Irons was approved after being amended to specify a $\frac{5}{8}$-in. length of thread, and a 1-in. length of stud.

The Springs Division reports on Spring-Eye Bushings and Frame Brackets for Springs were approved as printed. The report on Definitions was referred back to the Division because of criticism of the method of defining deflection, load height and free height.

The Lubricants Division's progress report that was presented only to secure suggestions and information for the Division's guidance followed a meeting held by the oil producing and consuming interests during the morning. A number of valuable suggestions were received and a revised tentative report will be prepared and circulated by the Division during the Summer.

A progress report was made by the Chairmen of the Passenger-Car and the Engine Divisions on their study of methods of numbering engines and frames for theft prevention, and to secure reduction of automobile theft-insurance premiums. The report was supplemented by the exhibition of a number of models and by lantern slides illustrating the application of many methods that have been considered. The progress reports on Metric Thrust Ball-Bearings, Brake-Lining, and Starting and Lighting Equipment were not given on account of the lack of time.

R. M. Hudson, of the Division of Simplified Practice of the Department of Commerce, presented a very interesting paper on the work of that Division. He explained the service that it is felt the Department of Commerce can render the automotive industry and the public through organized cooperation with the Society of Automotive Engineers and the National Automobile Chamber of Commerce. Mr. Hudson's paper will be printed in a later issue of THE JOURNAL.

The action taken by the Standards Committee on the reports submitted by the Divisions was reported to and approved by the Council and at the Business Session of the Society held Tuesday evening. The reports that were approved will be set forth in the August issue of THE JOURNAL, and submitted to a letter ballot of all the voting members of the Society.

### NOMINATION OF 1923 OFFICERS

H. W. Alden was nominated to serve as President of the Society for the next calendar year by the Nominating Committee, which was completed and organized at the White Sulphur Springs Meeting. The committee reported the following other consenting nominees for the elective offices next falling vacant under the constitution, i.e., after the 1923 Annual Meeting of the Society:

First Vice-President—H. M. Crane
Second Vice-President, representing motor-car engineering—(Undecided)
Second Vice-President, representing tractor engineering—A. W. Scarratt
Second Vice-President, representing aeronautic engineering—E. P. Warner
Second Vice-President, representing marine engineering—E. J. Hall
Second Vice-President, representing stationary internal-combustion engineering—(Undecided)
Councilors (to serve during 1923 and 1924)—W. A. Chryst, F. W. Gurney and A. J. Scaife
Councilor (to serve during 1923)—H. M. Swetland
Treasurer—C. B. Whittelsey

The members of the 1922 Council who will hold over during 1923 are B. B. Bachman as past-president and Councilors C. F. Scott and L. R. Smith.

The Nominating Committee was constituted of Cornelius T. Myers (chairman), Metropolitan Section; V. G. Apple, Dayton Section; H. R. Corse, Buffalo Section; T. F. Cullen, Pennsylvania Section; L. A. Emerson, Minneapolis Section; W. S. James, Washington Section; T. J. Litle, Jr., Detroit Section; R. J. Nightingale, Cleveland Section; B. S. Pfeiffer (secretary), Mid-West Section; L. W. Rosenthal, New England Section; M. A. Smith, Indiana Section; and V. E. Clark, F. S. Duesenberg and G. E. Goddard, members-at-large. This was the annual Nominating Committee, provided for by the Society's Constitution, under which 20 or more members entitled to vote may constitute themselves a special Nominating Committee, with the same power as the annual Nominating Committee. The By-Laws of the Society provide that a special Nominating Committee, if organized, shall on or before Nov. 15 present to the Secretary of the Society the names of the candidates nominated by it for the elective offices next falling vacant, together with the written consent of each.

### BUSINESS SESSION

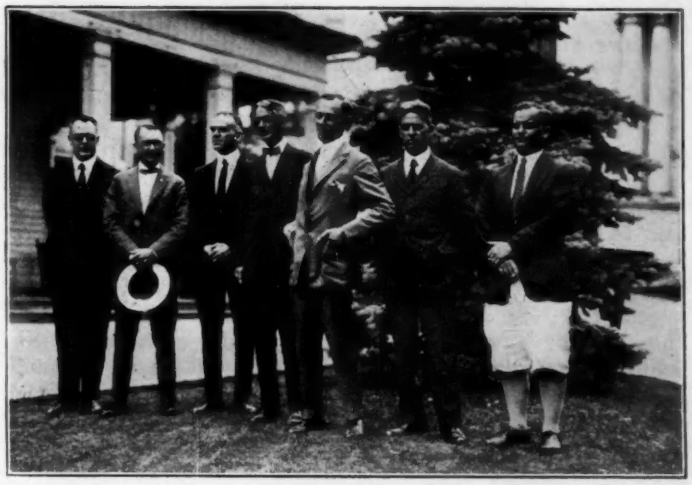
President Bachman's address, which is printed in full elsewhere in this issue of THE JOURNAL, was received very cordially at the Business Session held Tuesday evening.

It was reported that the Society's net loss for the first 8 months of the current fiscal year was $18,360.26. During the corresponding period of the last fiscal year the Society had an unexpended income of $14,504.88. The loss in income this year is due to a reduction in receipts of $27,439.30 as compared with last year. The amount of initiation fees from new members was $6,335.00 less than

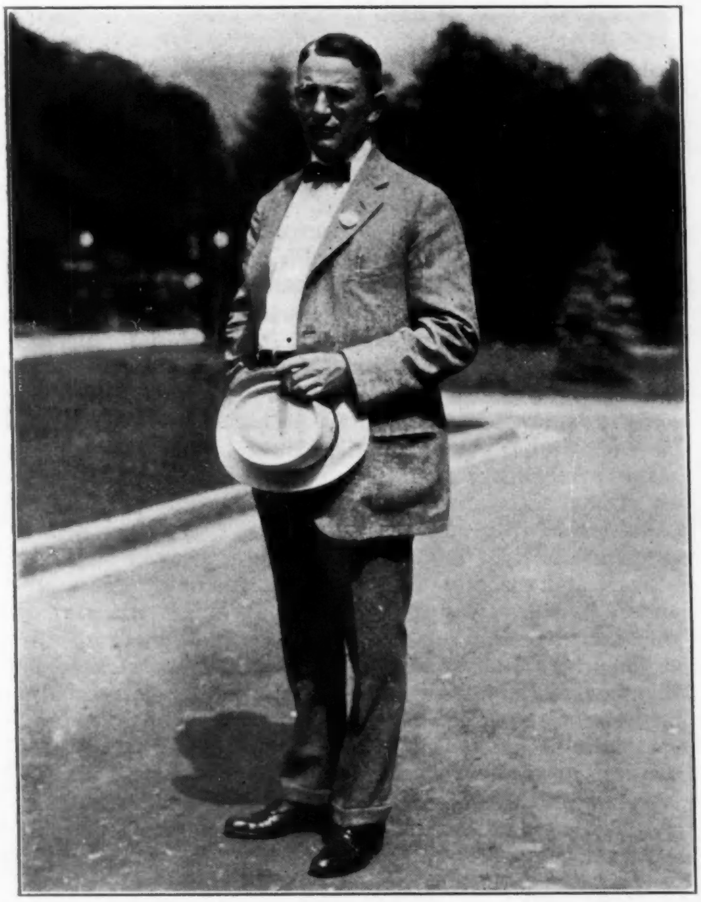
for the same 8 months of the last fiscal year. As a result of careful management the total operating expense for the period was increased only $5,425.84, notwithstanding added activities involving expenditure of approximately $10,000.

On April 30 the assets of the Society amounted to $180,127.63, these being offset by accounts payable of $9,159.29 and special reserves of $50,758.11; leaving net assets of $120,210.23, approximately $90,000 of this amount being in the form of United States Government and railroad securities.

The Meetings Committee announced that a two-day national Society meeting devoted to production engineering matters will be held in Detroit during October, this meeting including technical sessions and factory inspection trips. It was stated also that the annual Service and Tractor Meetings would be continued next year.



Some of the Council



Herbert W. Alden

The Membership Committee reported that the total enrollment of the Society on May 31 of this year was 132 more than on the corresponding date of last year, notwithstanding the fact that several hundred members were dropped for non-payment of dues or other causes.

The Sections Committee announced that all of the Sections are in a healthy financial condition and that many excellent papers had been presented and discussed at their meetings during last season. The committee advised strongly the practice of the Sections arranging their programs for the year in accordance with pre-determined plans. The Sections luncheon held on Thursday is reported at some length elsewhere in this issue of The Journal.

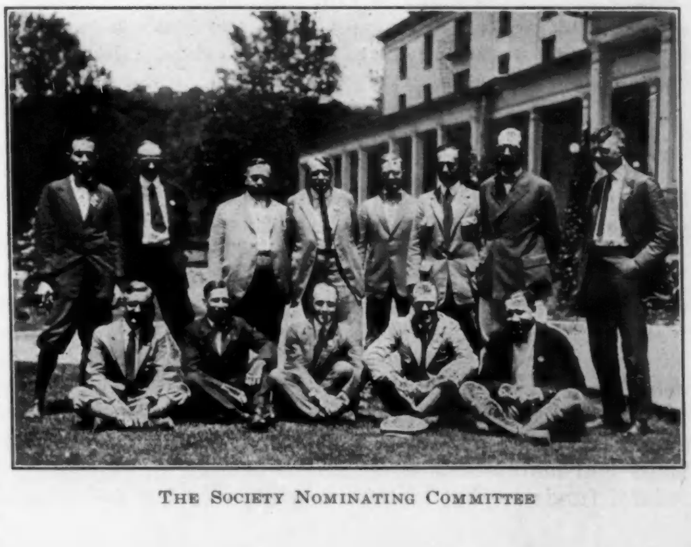
Under the item of new business, a lengthy discussion was had on the Society's affairs in general, including the matter of the grading of applicants for membership, Sections activities and the holding of local meetings in cities at which no Sections of the Society are located.

### Highway Matters

Director W. K. Hatt, of the Advisory Board on Highway Research of the National Research Council, presented at the Friday morning technical session a valuable uptodate survey of studies on highway matters that are being made by different institutions throughout the Country. Professor Hatt later conferred with H. W. Alden, chairman, G. A. Green and Prof. W. E. Lay, of the Highways Committee of the Society, with a view to coordinating further the technical efforts of highway and automotive engineers. The Highways Committee of the Society, the Highways Committee of the National Automobile Chamber of Commerce and representatives of the Association of State Highway Officials are scheduled to hold joint sessions this month.

### Simplification of Practice Through Department of Commerce

At the instance of the Division of Simplified Practice of the Department of Commerce, the National Automobile Chamber of Commerce and the Society of Automotive Engineers have appointed committees to make specific recommendations in furtherance of wider reduction to practice of automotive standards established and to be established. The National Automobile Chamber of Commerce Committee is constituted as follows: F. E. Mos-



The Society Nominating Committee

kovics, chairman, D. C. Fenner, H. B. Harper, M. L. Pulcher and C. E. Salisbury. The personnel of the Society's committee is J. G. Vincent, chairman, C. F. Kettering, C. M. Manly, F. E. Moskovics and W. G. Wall.

These committees held sessions last month at New York City and at White Sulphur Springs during the time of the Summer Meeting of the Society. The purpose of the Department of Commerce was thoroughly endorsed and the economic value of the proposed procedure fully appreciated. It is believed that the joint action will result in wider use of standards and the reduction of unnecessarily long lists of various sizes and sets of dimensions in the automotive field. It is the opinion of the committees that the first subjects to be taken up and studied are ball and roller bearings, solid and pneumatic tires, starting and lighting batteries and their carriers, and certain spark-plug dimensions that are essential for most satisfactory vehicle maintenance by users.

The facilities of the Society will be utilized in collecting data and making studies in connection with the work of the two committees and the cooperation of all allied associations concerned in the industry will be sought.

### The Aeronautic Session

Unfortunately the attendance of engineers interested in aeronautics was very small at the White Sulphur Springs meeting. Those who did attend showed great interest in the papers of Prof. E. P. Warner and Capt. G. E. A. Hallett, both of which were printed in the June JOURNAL. Capt. Hallett's paper was abstracted by E. T. Jones, the author being unable to attend the meeting due to illness. Mr. Jones supplemented the paper with entertaining motion-pictures depicting some of the engineering and flying activities at McCook Field. These were followed by authentic aviation and front-line action pictures of the recent war with Germany, which were loaned through the courtesy of the photographic section of the Signal Corps. The paper by C. L. Egtvedt was read by title only in the absence of the author.

V. E. Clark, chairman of the session, asked Professor Warner whether he had considered including some measure of the wing-curve characteristic such as $K_{y\ max.}$ in his performance formula and if this would not assure greater accuracy in the results obtained. Professor Warner stated that this addition would tend to destroy the simplicity of the suggested formula without increasing its accuracy appreciably. H. M. Crane congratulated Professor Warner on the derivation of a formula, devoid of unwieldy mathematics, that gave results sufficiently accurate for design approximations. In answer to a question, Professor Warner stated that parasite resistance is not a factor in determining the ceiling of an airplane because the greatly increased angle of attack at high altitudes results in a very large wing-drag which dwarfs the parasite resistance. However, at high speeds and a small angle of incidence, parasite resistance is a limiting factor since the drag is then relatively small. After the presentation of Captain Hallett's paper there was a general discussion commending the work of the Air Service Engineering Division at McCook Field. Professor Warner considered it inadvisable to divert all government funds to industrial plants for development work, although efforts were being made to bring about this condition. He favored leaving the work of a research character in governmental laboratories since the results would be circulated generally for the benefit of all in the art. H. M. Crane considered that Major T. H. Bane had maintained an excellent balance in expending federal funds so that his own engineers and those in the industry could be employed to the best advantage. He favored strongly the continuous testing and development of engines and planes of accepted design, believing that our most successful aircraft are the result of slow and rigorous development work. Elmer A. Sperry believed that McCook Field was of great assistance to the industrial designer because of the service it rendered as a source of research information. He strongly recommended a continuance of ample appropriations for the Field's use, feeling that the money could not be expended more efficiently. Professor Warner and Mr. Crane both argued for a closer contact between McCook Field and the airplane designer after the experimental models had been accepted and started through their flight and performance tests, this being the period when they were undergoing alterations to attain a final state of development.

### Research Session

H. M. Crane, chairman of the Research Committee, called the meeting to order and explained the change that had been made in the program. The first paper on the program, that by W. S. James on Fuel Research at the Bureau of Standards, was deferred until Saturday morning, and F. C. Mock read the paper prepared by him in collaboration with M. E. Chandler on the Hot-Spot Method of Heavy-Fuel Preparation. In connection with the paper by Mr. James, Mr. Crane brought out the importance of research work on the fuel-consumption in automobiles. The question of fuel in automobiles on the road has been a matter of so many miles per gallon, with an occasional explanation of the kind of service undertaken. Mr. James's study at the Bureau of Standards was undertaken with the idea of stating this problem in other terms. Instead of miles per gallon, his ambition was to determine inches per drop, or something to that effect. A complete set of new apparatus was designed to enable the Bureau to determine instantaneous fuel-consumption over almost any kind of trip. The Society was so impressed with the interest and value of this work that it was voted to support it in every way possible, and to make it a part of the fuel investigation undertaken by the Research Department. The cooperation of the National Automobile Chamber of Commerce and the American Petroleum Institute was secured, both agreeing to supply money, material and labor. The tests, when completed, will give us a more complete idea than we ever had before, of the way in which gasoline is actually used in road service.

C. T. Coleman read a paper describing the wholesale use of fuel of different grades in fleets of Government trucks, operated in normal service, and the results obtained from the observation of mileage and other characteristics. Post-Office trucks were used, operated by an exceptionally high-class personnel, under the best economic conditions. Four fuels were used, lettered *A*, *B*, *C* and *D*. The second, which was the key fuel, represented the average standard commercial gasoline that we are buying today at the service-stations. Tests were run in Philadelphia and Pittsburgh, the first city representing smooth going with few hills and grades, and the second hilly country and cobblestone streets with considerable grades. In Philadelphia, 37 3/8-ton Fords, 83 General Motors 3/4-ton trucks, 31 General Motors 1 1/2-ton trucks and 12 3-ton Packards were used. In Pittsburgh there were used 42 White 3/4-ton trucks, 16 2-ton Whites, and 16 3-ton Rikers. Every type of truck represented a different carbureter and a different kind of induction system.

The different groups were divided equally in respect to

kind of trucks, tonnage, and service performed. One section was run on the key fuel *B* at all times, and the other section on *A*, *C* and *D* fuels, thus enabling the testing engineers to compare the two units of trucks, or the one fuel with the other. Fuels were changed every week. The mileage was noted when the gasoline was put in, and every time the fuel was changed the residue of fuel was drained out and weighed. At the end of each week or of each run, a quart sample of the crankcase oil was taken and sent to the Bureau of Standards for testing.

Taking the distillation curves of the fuels used, *A*, *B*, *C* and *D*, we might say, taking *B* as the key fuel, to represent 100-per cent production per barrel of crude oil, *A* represents 80.9-per cent production per barrel; *C* 114.4 per cent; and *D* 127.8 per cent.

Slides were shown, giving average results for different trucks run in Philadelphia and Pittsburgh, resolved into percentages of miles per gallon. *The outcome of the investigation was, there was very little difference in the percentage of miles per gallon of the different fuels, but there was a considerable difference in crankcase-oil dilution on these fuels.* In all the trucks there was a slight advantage in favor of the *A* fuel, and it was noticed that with a fuel of heavier quality or of higher end-point, the crankcase dilution tended to increase. There was more crankcase dilution on *C*, and proportionately still more on *D*. Slides were thrown on the screen giving the curve of the percentage of dilution against volatility.

Summing up, Mr. Coleman said, taking the percentage of miles run on the *B* fuel per barrel of crude as 100, we find that fuel *A* gives 86-per cent miles per barrel, and 3.4-per cent actual crankcase-oil dilution. With *B* fuel the crankcase dilution is 4.7 per cent. With *C* fuel, representing 123-per cent miles per barrel of crude, the figure for crankcase dilution is 6.8, and with the *D* fuel, averaging 136-per cent miles per barrel of crude, the crankcase dilution is as high as 9.1. It appears, therefore, that more miles are obtainable per barrel of crude on the heavier fuel, but here the problem of crankcase-oil dilution becomes more acute.

In opening the discussion, Mr. Crane urged the necessity of maintaining a reasonable degree of uniformity in the quality of the fuel, and advocated that the quality of the fuel used be adapted to the atmospheric conditions prevailing in different parts of the Country and at different seasons of the year. He commented on the fact that in Philadelphia, where the trucks were operating at part throttle, and with a fair degree of heat, the heavier fuels did not affect the mileage operation to a great extent. In Pittsburgh, however, where there was some real full-load operation, the effect of the increased end-point was much more noticeable.

P. S. Tice asked what apparatus was used to determine crankcase-oil dilution. Mr. Coleman replied that the values given were obtained in a 100-cc. Engler-flask, the apparatus used being a conventional distillation outfit with an ice bath and an ordinary Engler-flask. Mr. Tice considered that the "ordinary Engler-flask" was the weak point. In response to a query from F. C. Mock, Mr. Coleman explained that all carbureter adjustments were left to the mechanical force, since the test was carried out on an ordinary service basis. J. H. Hunt commented on the influence of cold starts and stops on crankcase dilution, and asked if any steps had been taken to correct for this. Mr. Coleman said that starts and stops were not taken into consideration because the trucks do the same thing every day, and it was simply a standard test, run under normal service conditions. Weather conditions were eliminated by the expedient adopted of running half the trucks on *B* fuel and the other half on the other grades of fuel.

Mr. Crane made some interesting observations on the importance of the human equation in tests of this kind, and added that the chief defect in laboratory and factory tests is that the mechanics are more skilled than the average. For this reason he contended that a test of this kind could afford no criterion unless it were extended over a considerable period of time. O. C. Berry argued that the use of the same fuels in the hands of the general public would not result in anywhere near the same mileage per gallon, and expressed the opinion that any test reporting miles per gallon should be preceded by a very careful series of performance tests to determine the correct carbureter setting. In thanking Professor Berry for his constructive criticisms, Mr. Crane urged that any other suggestions as to forms of test that might be considered helpful or desirable be sent to Dr. H. C. Dickinson, the research manager of the Society. Some interesting points were raised by P. J. Dasey who held that performance is the factor of chief importance in any test, while economy is secondary. He said that in his own laboratory he makes comparative tests for crankcase dilution using city illuminating gas instead of gasoline, and yet light ends may be found in the lubricating oil in the crankcase, which are not due to dilution, but to the cracking process that is constantly carried on in the cylinder. The dilution with heavy ends that is found in the crankcase are not the fault of the refiners so much as of the men who design the engines to handle the fuel. In reply to Mr. Dasey's suggestion that only one brand of oil be used in all such tests, Mr. Coleman stated that owing to the dissimilar oiling systems of the Ford and the other light $\frac{3}{4}$-ton truck, the same oil could not be used in both.

R. E. Carlson, of the Bureau of Standards, followed with a paper in which he gave a brief outline of the status of the cooperative fuel-research program that the Bureau of Standards has undertaken in cooperation with the National Automobile Chamber of Commerce and the American Petroleum Institute. He began by saying that the work may be called a continuation of that which Mr. Coleman has been doing on a somewhat larger scale, except that it is being conducted along more nearly laboratory lines. Four grades of fuel are to be used. The object of the tests is to determine the fuel-consumption under average road conditions with average cars in the hands of average drivers. These averages are rather difficult to determine but a program has been worked out, and the work is to be begun early this month. Mr. Carlson invited the cooperation of the members to determine what the program should be. The fuel is to be the same as that used by Mr. Coleman in his tests, but it is hoped that more accurate results will be obtained through the control of the various factors.

Mr. Mock presented a summary of the paper on The Hot-Spot Method of Heavy-Fuel Preparation. The authors investigated the various methods of heat application in an endeavor to produce the minimum temperature necessary for a dry mixture. It is their conviction that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold.

They found that the minimum temperature varied with the method of application of the heat, and proceeded to make an analysis of the available methods on a functional rather than a structural basis. Three of these methods are discussed

(1) When the heat from the walls of the manifold is applied through the medium of the air
(2) When it is applied to the fuel alone, or partly to the fuel and partly to the air
(3) When a spray of atomized fuel and air is directed against a heated surface

A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, could be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made, and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge, and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder, and not from the carbureter to the manifold. The paper was illustrated with drawings.

H. W. Alden opened the discussion by commenting on the mystery that has always surrounded the hot-spot. He called on Thomas Midgley, Jr., who described a new preparation brought out by the General Motors Research Laboratory to prevent fuel knock. He described it as tetra-ethyl lead, and offered to give a sample of it to anyone who cared to investigate its properties.

Dr. H. C. Dickinson then presented his paper, Progress of the Research Department. In it he emphasized the necessity for research, touched on the importance of the universities as bases of operation for pure research, described the resources of the Department and the special facilities that it has to offer and gave a short description of the work done in connection with the information service. He laid particular stress on the fact that the Research Department is equipped to give assistance to engineers and others in the industry and that all its resources are at their command. He concluded by describing the three major research enterprises in which the Society is engaged at the present time, namely the cooperative fuel research that is being carried on under the direction of the Bureau of Standards, the Society's fuel-research program, and the cooperative Highway Research program under the auspices of the National Research Council.

In discussing Dr. Dickinson's statement of the work done by the Research Department along fuel lines, P. S. Tice brought out the point that we have not yet arrived at a method of carburetion or handling of the fuel in the intake, that will give us the maximum fuel-economy, and suggested that we are wasting time when we try using several different fuels of known volatility in several different devices. T. J. Lítle, Jr., expressed the opinion that the most important task is to prepare for the heavier fuels that are bound to come. Chairman Crane pointed out that this work is to be accomplished by the present investigation, which will give us a curve that will indicate the utilization value of petroleum distillates of different volatility in engines of the present general type. Either the fuel must be modified to suit the needs of the present type of engine, or the present type of engine must be completely changed. He placed his confidence in the possibility of modifying the fuel to suit the engine that has been developed because of its simplicity and service, rather than in adapting the engine and complicating it to use it with some arbitrary form of fuel. There are millions of cars in service today which cannot be altered materially.

Dr. Dickinson explained that the tests, so far, had shown that the cars and trucks had actually utilized the heavy 500-deg. end-point fuel without any very marked decrease in economy. This does not take into account the question of crankcase-oil dilution. W. S. James alluded to the question now before the Federal Specifications Board as to the end-point of the most suitable gasoline. The refiners maintain that the present type of commercial gasoline is satisfactory and that the end-point or 95-per cent point can be raised without detriment. This matter is to be taken up by the refiners at a meeting to be held early this month. At the present time there are practically no data on the advantages or disadvantages in actual service of fuels with varying end-points. The few tests reported furnish at least some indication of whether the refiners' demands should be granted. He suggested that possibly the greatest gains could be expected from carbureter and car adjustments, rather than from a change in the fuel. Mr. Lítle said he believed that he expressed the view of the automotive industry when he stated that a better gasoline costing a few cents more would be eagerly welcomed. Chairman Crane endorsed this statement and added that the owner of the car is as much interested in the mileage he gets per gallon of fuel, as in the cost of the fuel per gallon. T. A. Peck stated that the petroleum industry was ready to go more than half way to meet the automotive industry, but that careful conservation of the existing supply of raw material must be the primary consideration.

### Fuel and Engine Session

Chairman O. C. Berry opened the session by calling on W. S. James, of the Bureau of Standards, to present his paper on Fuel-Volatility Research at the Bureau of Standards, with demonstration of test-car equipment. Mr. James's talk was profusely illustrated with lantern slides which showed the various devices that have been adopted by the Bureau in connection with its fuel-research program. Among the devices illustrated and described briefly by Mr. James were the accelerometer, the velocity-recording instrument, the oil and carbureter tester, draw-bar dynamometer, and a multiple recorder.

M. C. Horine inquired what data Mr. James had on the effect of friction of the springs. Mr. James replied that in driving at a uniform speed along level stretches of road it is possible to get a zero independent of spring action. Chairman Berry discussed calibration on level roads, and noted that inertia effect is eliminated largely because there is very little movement of the mercury itself in the accelerometer. The effect of temperature on the accelerometer can be practically eliminated by making the free ends wide.

Mr. Chase was inclined to think that inaccuracies would creep in owing to the instability of the pens due to irregularities in the road. Mr. James replied that the friction of the pens made very little difference owing to the motion of the paper and the vibration of the car. The pens do not stick, thus avoiding the resulting reduction of sensitivity.

Thomas Midgley, Jr., presented the paper, prepared in collaboration with T. A. Boyd, on Detonation Characteristics of Some Blended Motor-Fuels. The authors have measured the effects of admixtures of various percentages of alcohol and alcohol-benzene mixtures for reducing the detonating tendency of paraffin hydrocarbons. These results represent an extension of previous work in which similar determinations were made for benzene

and other aromatic hydrocarbons. The bouncing-pin apparatus was used for making the determinations. The data obtained by its use are considered to be remarkably accurate.

In order that the effects of the blending materials might be measured through as wide a range as practicable, they were blended with kerosene for making the majority of the determinations. This made it possible to ascertain the characteristics of the materials up to a concentration of 80 per cent of benzene or 50 per cent of alcohol without introducing the difficulties due to excessively high engine-compression. Because xylidine has the property of exerting a powerful suppressing action on detonation when present in a fuel in percentages that are relatively very small, the standard used as a basis of comparison in the tests was composed of small percentages of xylidine in the paraffin fuel. Tables and curves are appended that show the results of the tests in detail.

Mr. Bachman, in discussing the paper, asked Mr. Midgely if the kerosene dilution with benzol had proved more effective than high-test gasoline dilution with benzol. Mr. Midgely replied that his observations had borne out this surmise. Mr. Chase asked what procedure had been followed in setting the spark for measuring detonation. Mr. Midgely said that the spark had been set for the maximum power.

G. A. Round read an abstract of his paper on Oil-Pumping. He defined oil-pumping and mentioned its results. The influence of various operating conditions was brought out, particular reference being made to passenger-car service. The factors that control the rate of oil consumption were brought out, and some unusual conditions reported. Various features of piston grooving and piston-ring design were mentioned and the effect of changes illustrated. The relative advantages of the splash and force-feed systems as affecting the development of oil-pumping troubles were set forth and improvements suggested. A new device for reducing oil-pumping and dilution trouble was described and illustrated.

F. F. Kishline, discussing Mr. Round's paper, asked if the tests referred to were made on a dynamometer or on the road. Mr. Round stated that the tests were made on the road. In discussing methods adopted for dealing with an excessive consumption of oil, Mr. Round stated that oil has to be controlled at the source and cannot be regulated by a piston.

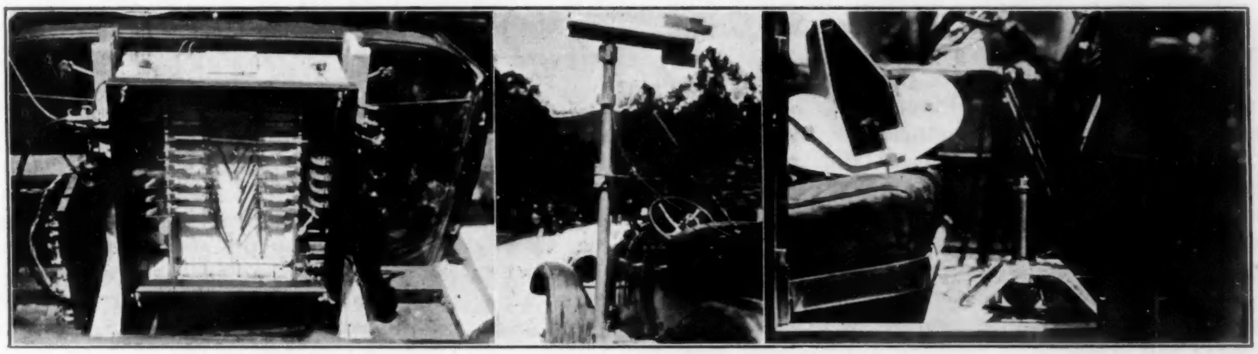
A. A. Bull gave his paper on Oil Consumption, in which he considered fundamental factors. No attempt was made to determine the differences between lubricating systems. Beyond the fact that different oils apparently affect the oil consumption and that there is a definite relation between the viscosity and the oil consumption, the effect of the physical characteristics, or the quality of the oil, did not receive particular attention.

A Portion of the Golf Course

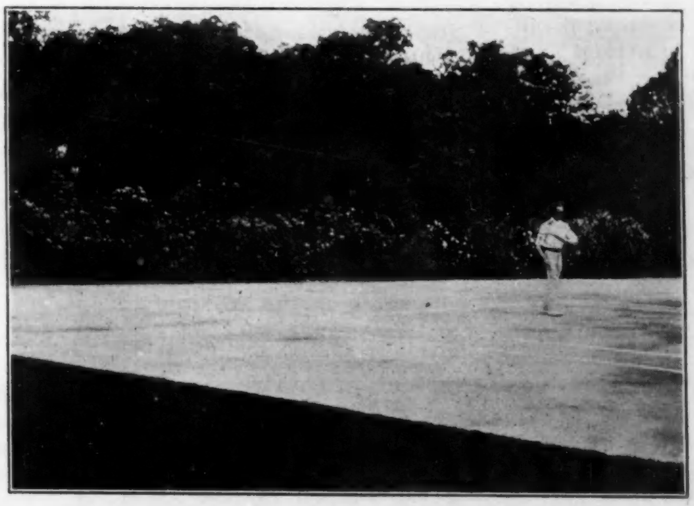
The paper describes the methods of testing and the subject is divided into

(1) The controlling influence of the pistons, rings and cylinders
(2) The controlling influence of the source from which the oil is delivered to the cylinder wall

The subject is treated under headings that include the piston-ring; the effects of oil-return holes, side-clearance and ring-motion; thin rings; influence of piston-fit; efficiency of the scraper-ring; ring and cylinder contact; carbonization and spark-plug fouling; oil-supply control;



Some of the Apparatus Developed by the Bureau of Standards for Testing the Performance of Engines on the Road



President Bachman in Action on the Tennis Courts

influence of oil viscosity; effects of dilution; external oil-leaks and breather discharge; and influence of controlling lubrication in proportion to throttle opening.

### Motorbus Session

President Bachman opened the Motorbus Session by introducing G. A. Green, vice-president and general manager of the Fifth Avenue Coach Co., who then read his paper entitled, Principles of Motorbus Design and Operation. The paper dealt with the fundamental characteristics of buses, the principles on which their design and operation should be based, fares, service requiremens and the unwisdom of overloading. The subject was treated impersonally, except for specific references that were made from time to time to the practice followed by the Fifth Avenue Coach Co. in its equipment.

The factors controlling bus design are said to be

(1) Safety
(2) Comfort and convenience of the public
(3) Minimum operating cost

The various subdivisions of each were commented on in some detail and numerous illustrations and tabular data supplemented the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are incapable of rendering satisfactory service as buses, and that if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

In opening the discussion, President Bachman emphasized three points as being of particular interest, namely, powerplant construction, control and stability.

R. E. Plimpton then read his paper on Some Fundamental Characteristics of Present-Day Buses. He enumerated the distinctive features of buses designed for city, inter-city and rural services, illustrating his remarks by practical examples. Steam and electric motive-power were discussed and considerable attention was given to chassis components requisite for bus service. The general types of bus body were treated, together with the influence of climate and local preferences.

Mr. Plimpton discussed the comfort and convenience of passengers and problems of heating, lighting and ventilation and methods of fare collection. State and local regulations were referred to in connection with their effect upon bus operation. The paper was accompanied by illustrations and a table showing condensed specifications.

President Bachman opened the discussion by remarking that Mr. Plimpton's paper represented the technical newspaperman's point of view on the bus problem, as distinct from the preceding paper that treated the subject from the engineer's standpoint. Mr. Green agreed with Mr. Plimpton that one of the greatest problems from the viewpoint of both designers and operators of motor buses is the elimination of all unnecessary parts. Aside from the matter of weight and first cost, there is the question of repairs and renewals, also possible failures and consequent delays and service interferences. For normal bus service he advocated a four-cylinder engine of approximately 300-cu. in. capacity. Referring to Mr. Plimpton's comments on the possible advantages of the six-wheel bus, Mr. Green said he believed this design involves additional parts, weight and cost, and that the value of this form of construction had not been proved. He said emphatically that in his opinion the six-wheel vehicle should be considered as being still in the early experimental stages. H. L. Howell gave an interesting description of the types of bus operated by the London General Omnibus Co. Before the war the company used the B-type bus, with a 34-passenger seating-capacity. This was succeeded during the war, when the buses were used for transport service on the Western Front, by the K-type, 46-passenger bus, which in its turn has been succeeded by the S-type 54-passenger bus. He added that the cost of operating the S-type 54-passenger bus is not much greater than that of the B-type 34-passenger bus.

Mr. Green brought out an interesting point in discussing the types of equipment necessary for bus service. He stated he believed that in the majority of instances, initial operation ought to be commenced with single-deck vehicles; then after the service is built up, double-deck buses should be added; and that practically all operations require the two distinctive types. After the service has reached a point where double-deck vehicles are required, the single-deck type will be found extremely valuable to aid in the natural process of development for operation during cold and wet weather and for all kinds of special service.

Several members inquired as to what extent automobile companies are specializing on the production of motorbuses, since it is evident that the motorbus requires special parts and specifications. Mr. Plimpton said that several companies are now specializing on such construction. It is hoped that in time the building of motorbuses will be so simplified and standardized that it will be possible for any operator to order a fleet of buses that will be standard for the type of service he has in mind. Here again, the need for standardization work by the Society was strongly emphasized.

### Passenger-Car Session

J. V. Whitbeck, chairman, introduced P. M. Heldt, who read his paper on Overhead-Camshaft Passenger-Car Engines. The paper illustrated the steady increase in the use of overhead valves in passenger-car engines from 6 per cent in 1914 to 31 per cent in 1922. This increase has been largely effected by the successful operation of overhead valves in aircraft engines and by the publicity given this type of valve through its almost universal adoption on racing machines. Mr. Heldt described the methods of operating valves in the cylinder-head; the advantages of the valve-in-head construction as regards the form of combustion space, engine cooling and high-speed operation; the reason for using an overhead camshaft to operate the valves on racing engines, the question of noisy operation and the possibility of having an overhead-camshaft engine operate as quietly as one in which the camshaft is enclosed in the crankcase; the feasibility of silent operation with a rear drive; the use of the various types of gear for the camshaft drive, with an estimate of the advantages and disadvantages of each, illustrated by practical examples; and some radical designs of overhead-camshaft drive and valve-actuating mechanism developed abroad.

Mr. Crane opened the discussion by commenting on the causes of noise, which he attributed to the seating of the valve. He stated that the valve opening is almost noiseless. In comparing the relative merits of the different types of engine, he stated that the overhead-camshaft engine lacks accessibility and the drive tends to be noisy. For a quiet high-speed engine he considered the L-head most suitable. For a moderate-speed engine he preferred the overhead-valve construction, which can be made almost noiseless.

(*Concluded on page* 119)

# Presidential Address of B. B. Bachman

THE activities of the Society during the period since we met in New York City have been numerous. I would like to lay a few of them before you in the way of a report, covering in a general way the problems that have come before the Council. This can be accomplished best by following the work as divided among the administrative committees.

### The Meetings Committee

The results of the activities of the Meetings Committee during this period need little comment as the material evidences are before you, and will unfold during the session in which we are now participating. On the basis of the work that I know they have each done individually, I believe that I can anticipate your unanimous approval, and extend to the Committee your appreciation as well as that of the Council for the sacrifice of time and effort that they have made.

For the future, I know the Committee is interested in the activities of the Sections, of which I will speak more in detail later, and will welcome the opportunity for more active cooperation with the Sections in establishing a rounded-out and harmonious program of meetings for the year. As a large part of the value of the Society to its members lies in the meetings that are held, there has been some discussion as to whether it would not be wise to hold annually more than the two meetings for which the Constitution provides, and very definite thought is being given to this subject.

### Society Membership

The Membership Committee has been dealing with a very serious problem in the affairs of the Society. The growing activities and the new fields of work that are opening around us on every side require the addition to our ranks of all who are equipped to assist us in our work or who can be benefited by it. While the scope of our organization is large, it is in a certain sense, limited. We must keep in mind that restricting our membership to the field of active designers of automotive vehicles and their more important component parts would be a very narrow policy. On the other hand, we must recognize that opening our doors for the general admission of all without regard to the service we can render them or they can render us would be unwise. Between these two extremes, the Committee and your Council believe, there is a sufficiently large field for the Society to draw from in permitting a rational and satisfactory growth.

This field, it seems to me, will cover, first, the designer of automotive apparatus and their important component parts; the instructor in arts and sciences relating thereto; the specialist, or the man versed in the design, construction and efficient operation of the production agencies for materials and completed structures; and, finally, but by no means least, the man who is skilled and is competent by training and experience in the maintenance and operation of the apparatus that the industry manufactures. This view may be criticized as being too broad, but I am firmly convinced that unless we recognize the valuable assistance that these other classes can render in developing the field of internal-combustion power application, and are willing to receive them on the basis of equality to which their ability and dignity entitles them, we will be retarding our attainment to that position of consideration to which we are entitled.

In the class of those who, while not engineers in the broad sense outlined above, are nevertheless interested in the work that we are doing, and directly or indirectly are benfited by it, there should be included with the man engaged in marketing our products, the man who has the responsibility of purchasing materials that we use in our processes of manufacture. Both these classes can in many instances give a wider perspective to our vision and can be materially benefited by our activities.

While the continuing growth and activity of the Society require careful consideration of our financial resources, and while the fees that are received from our members are a very important source of revenue, we must guard against any plan of membership increase that has only revenue in mind. The work that the Society is doing and the service that we can render to each other as members cannot be measured by the cost, nor should we knowingly solicit membership for mere financial support from anyone.

Another problem that the Membership Committee has actively in mind is the question of retaining the active interest of those who are already members. The recent industrial conditions have brought about the reduction of organizations which has resulted in personal difficulty for some of our members. There are others who possibly have been persuaded to join on the wrong basis, and have therefore never really been members.

In any event, whatever the reason may be, it is probably only natural that there should be a percentage who are delinquent in their financial obligations and thereby, as a result of our constitutional provisions, deprived of some of the benefits of membership. Extraordinary efforts have been made during the past months to increase the value and effectiveness of our employment service for the assistance and benefit of those who are numbered among those who have suffered personal loss through the recent unsettled conditions. As for the others, it is difficult to say just what steps should be taken to stimulate their interest and to retain their active association with us, but several methods are being actively canvassed toward this end by various committees and the Council.

### The Sections

The work of the Sections Committee in the phase of the Society's activities that it represents is undoubtedly of prime importance. As was to be expected, the proper organization of these activities has presented for a number of years some very intricate problems which are far from being settled, even today. Fundamentally, an organization of the character of ours is dependent in a large measure upon two things for its success; first, the character and value of its meetings; and second, the character and value of its publications.

It is manifestly impossible with a widespread membership that more than a relatively small percentage can attend meetings such as this and at the same time the multiplication of meetings on a national scale at more frequent intervals, while desirable in some ways, will not in itself fulfill all the possible functions of an active local Section. The theory upon which we have conducted the affairs of the Sections to date has been to give them a very large measure of indpendence, allowing them to direct and regulate their affairs in accordance with their local needs and under officers of their own selection.

They have been financed in large part by the contribu-

tions of the members of the Society who have joined these Sections, the Society contributing to the support of the Sections in addition. The reason that the present system has been adopted is that it has been felt that only a percentage of the membership of the Society would be so located geographically as to benefit by participating in Section activities, except to the extent that the multiplication of Section meetings with the presentation of valuable papers would furnish subject matter for THE JOURNAL. In view of this, it was felt that, even if it were possible, it would be wrong to appropriate from the funds of the Society the necessary amount to finance completely the Sections whose activities would be of direct benefit to only a portion of the membership. At the same time, on the basis of the increased contributions to THE JOURNAL, it was felt that some appropriation was justifiable.

Against this theory of the present method, we have the view that those members who wish to participate in Section activities should not have to pay further dues than those to which they obligate themselves in joining the Society; and it is urged that this additional taxation is in a large measure responsible for difficulty in bringing up the membership of the Sections to what it should be. There is much to be said for both of these viewpoints, but fundamentally I do not believe that in either one of them lies the secret of success or the reason for failure in the Section activities.

It has been suggested that a very large percentage of our members do not care to attend Section meetings, and unfortunately this is probably only too true; but I believe that one of the reasons for this lack of willingness to attend is that a proper survey of the need of the members has not been made and the programs that have been put forward have not been consistently of the caliber or kind to attract a consistent attendance. This statement is not made with any intention of criticizing any of the past or present Section administrations. It is merely a fact that I believe we must face thoroughly, and a problem that we must solve before we can put the Section work on the high plane where it should be.

In a large degree the Section problem is similar to the Society problem. While we recognize the sacrifice of time and the effort and thought contributed by committee members, we surely recognize that if it were not for our headquarters organization we would be lost. I know of no individual who has the ability for organization and the time to devote to the detail work of properly conducting a Section. The result is lack of continuity of effort and policy.

### SOCIETY FINANCES

During the last year, the question of finance has caused your officers considerable thought and anxiety. For the first time in many years, our current revenues have been insufficient to meet our expenses and afford a margin to be transferred to surplus. This is due to several things: first, to a loss in income from our advertising; and, second, to a reduction in the number of new members. Both of these conditions, it is believed, are temporary and will show improvement with a recovery of normal business conditions. We should nevertheless recognize that growth in usefulness and numbers will require not only continuation but expansion in our services, which will require thought and careful planning to balance the budget. It would have been possible this year to meet this emergency by a reduction in our activities, and it would be easy to recommend raising dues to prepare for the future. It is, however, the feeling of your officers that it is in times of commercial difficulty that organizations of the character of ours should increase rather than decrease their activities, for the reason that it is during times of this kind that the members individually and the industry as a whole need the greatest stimulus.

Regarding the raising of membership dues, there are of course many who could meet an increase with little difficulty, but on the other hand there are a number who, while they are vitally interested in the work of the Society, have found it impossible to meet the current dues. It has been my privilege to conduct a rather extensive correspondence with this smaller group. This has been, of course, in many instances of a confidential nature, but I am not violating that confidence in telling you that the opinion expressed above has resulted from this contact.

Nevertheless, we had to meet this problem. A survey indicated that we were spending a considerable amount of money annually in publications. While it is recognized that this is a legitimate and important function of the Society, conditions have changed; and the changes had not been reflected in the publications' policies. When the Society was first organized, it, in common with other engineering organizations, published its proceedings in the form of an annual or semi-annual volume containing complete papers and discussion of them. So long as there were no other or better avenues for the distribution of information to the members, this was very well. However, in later years, we have, through the activities of the Standards Committee, published the S.A.E. HAND-BOOK; later the *Bulletin*, which has developed into THE JOURNAL, was brought into existence. In THE JOURNAL we have a means of presenting to the members at a much earlier date than was possible in the TRANSACTIONS a complete record of the proceedings of the Society. Therefore, it seems that it would be highly inefficient for the Society to continue indefinitely to distribute the complete proceedings in THE JOURNAL and then at a later period duplicate this information in the form of a bound volume, without charge to the members in addition to dues.

There are many questions connected with this problem which it would be impossible to cover except in an inexcusably lengthy manner. Suffice it to say that, after long and careful consideration, the Council has finally decided that the TRANSACTIONS for the years 1921 and 1922 shall be sent to only those members who indicate that they wish to receive them. There will be no additional charge for these. After that time, it is the recommendation of your present Council, that the TRANSACTIONS be sold to the members at a nominal price which will partially cover the cost of production. This will permit several things: first, it will enable us to concentrate in greater degree on making THE JOURNAL more uptodate and complete in its record; and, second, it will relieve the finances of the Society of a burden that in a large degree under the old order was imposed for a service that was of little or no benefit to a large proportion of our membership. That this viewpoint is correct we believe is demonstrated by the fact that there were orders for only about 1200 copies of the last issue of the TRANSACTIONS.

I can appreciate thoroughly that a change of this nature will seem radical to some. I also appreciate the powerful influence of precedent and the fact that the receipt of bound volumes of TRANSACTIONS has long been a perquisite of members of engineering societies. On the other hand, this Society has in a large degree established itself and justified its existence as an organization on the basis of a disregard for precedent; and I

believe the other arguments that I have outlined herein are ample justification for the step that your Council has taken, and trust that the development of the plan will recommend itself to those of our loyal and interested members who have felt inclined to question the wisdom of the step.

### The Standards Work

With regard to the Standards Committee, I think you will recognize that, in view of my long association with this phase of our work, I am most vitally interested in what is being done. We were fortunate this year in being able to get a complete working organization of the Standards Committee going very promptly. A considerable amount of work has been done, as evidenced by the reports that were presented to the whole Committee by the Divisions this morning. This part of the work speaks for itself, and I will not do more than make this reference to it.

There are, however, certain other phases of the Standards work to which I wish to call your attention. At the risk of being tiresome, I would repeat what has been said so often before, that the Standards work is one of the most important activities of the Society. It has been suggested that the direct benefits of this work have reacted in favor of the industry as a whole, rather than of the individuals who hold membership in the Society and from whom we obtain financial support in large measure, and that, in view of this fact, it would be well if means could be found that would place the financial burden for the support of this work on the shoulders of those who most largely benefit from it. While there is no doubt as to the soundness of these suggestions from the viewpoint of placing the burden of expense on the shoulders of those to whom the benefits accrue, there are other considerations which should have our thoughtful attention. I am placing before you herein what are largely my own opinions, and hope that you will recognize them as such.

I believe that the fundamental strength of the Society resides in its being an association of individuals, and that the strength of our position as an impartial agency for the conduct of many of our activities would be jeopardized were we to make provision for corporate memberships the main purpose of which would be to obtain revenue. We have been fortunate in having our work recognized by several trade organizations which have indicated their approval and support by making financial contributions. The degree in which these have come to us, and representing as they do, not individuals but groups, I believe is therefore the most practical solution for this problem. It would be of considerable assistance and I believe perfectly proper if this form of recognition were extended. However, whether it shall be or not, we should exercise all our ability and energy to proceed in a rational way to extend and continue the Standards work that was inaugurated about 12 years ago, and has been carried on continually since. Cooperative endeavor of this sort, bringing together as it does the individual members of the Divisions, is of the greatest benefit in promoting the development of the individual and the building-up of the spirit of service that is vitally essential to the health and growth of such an organization as ours.

I wish that we could find some way of still further impressing upon the industry the importance of this work; that we could find a successful method of definitely determining the degree to which the S.A.E. Standards are used and a more definite measure of the economies that their use brings about. While it is necessary for the purposes of efficient organization that the Divisions be not too large, I would like to see the time arrive when the Division meetings should partake of the nature of technical sessions to which not only the Division members but all interested members would feel free to come and would desire to come. One particular reason for this feeling is my belief that as we grow and continue this work we must be more and more particular that the subjects proposed for standardization are properly considered and thoroughly analyzed in view of the broadest possible experience and opinion, so that assurance may be had that all interests have been properly represented. I wish that we could individually make it our plan and purpose to sell the idea that it is good business and money well invested for organizations to give their engineers the time and to assume their expenses in the attendance at these meetings as well as those of a more general nature.

As we proceed with the work of standardization, we will encounter more and more difference of opinion as to how far it should be carried. There are those among us whose breadth of vision carries them far in the list of subjects that they believe can rationally be standardized. There are others who feel that these suggestions if followed would be unwise, and would result in harmful restriction of initiative in the design and development of our apparatus. I hope that these two views will always be in evidence, but that they will be brought into contact in the work of the Committee, so that they may temper each other and produce a rational result. I believe that any student cannot but recognize that it is in the reaction of the extremes of opinion in their contact with each other that sound and conservative policies are formulated.

In connection with the work of the Standards Committee, it is well for us to recognize the increasing recognition of the importance of such work in every quarter. In the January issue of the *Automobile Engineer* Basil H. Joy outlined the work of seven committees, operating under the British Engineering Standards Association, having to do with the general subject of automobiles. These subcommittees are dealing with nomenclature, steel, small fittings, electrical fittings, wheels, rims and tires, and cast iron. You will recognize that for practically all of these our Standards Committee has already brought into existence valuable standards. The Department of Commerce, under Secretary Hoover, in the Division of Simplified Practice is taking a very active interest, as the name of the Division suggests, in the simplification that can be obtained by the adoption and use of standards. You have had the opportunity of hearing today from Mr. Hudson of the Division exactly what its aims and ideals are, and we hope in cooperation with the representatives of the National Automobile Chamber of Commerce to be able to further this work.

### Research

With regard to the work of the Research Committee, I feel that it would be presumptuous for me to attempt to make an extensive statement. We are devoting to this subject a session which, in conjunction with the report of Mr. Crane, chairman, and Dr. Dickinson, manager of the Department, will go farther than it would be possible for me to go, in outlining what has been done and what it is proposed to do.

My remarks on the work of the Standards Committee bear with equal force on the work of the Research Committee. The work in itself is largely of a character that will produce benefits that, in a considerable degree at least, will permit their being secured by others than the

individuals who comprise the membership of the Society. This viewpoint should not, however, blind us to the potential benefits that can accrue to the members. I say "potential" for the reason that the benefits will not be secured except insofar as the membership participates in the work and what each member will get out of it will, in a large degree, depend upon what he puts into it.

The results may not be startling in their scope at the present time, but I am firm in my conviction that the foundations that have been laid, if built upon with patience and with the consistent support of the membership, will in the very near future justify the inclusion of this work as a part of our regular program.

### New Developments

After this more or less hurried summary of the affairs of the Society, I would direct your attention to a more general survey, with a view of determining along what lines our activities as engineers and as an engineering society should be directed in the immediate future.

The period of industrial depression through which we have gone should be productive of some lessons to which it would be well for us to give thought. Naturally, those that appeal to me most forcibly and which I feel most competent to discuss are those having to do with the truck rather than the passenger vehicle. There have been three outstanding developments during recent months, the appearance of which may be due in part to conditions resulting from the depression. They are: the speed-wagon, the motorbus and the motor rail car. That there is a fertile field of usefulness for all three of these types can probably be accepted without question. That they each present features of design requirements which are distinctive and possibly not yet fairly appreciated in general is, I believe, also true.

We held in January and will hold at this meeting a session dealing in a degree with the problem of bus transportation. There have been sessions held by the Metropolitan and the Indiana Sections that had to do with the matter of the motor rail car. The problem of the speed-wagon may be more commercial than technical, but I believe that it deserves consideration. I am mentioning these points with the hope that our Sections will find some suggestions for their development for meeting topics.

### Highways

The question of highways is one that has been given considerable attention in the past in our discussions and should receive continuing attention. The ability and the efficiency of the vehicles that we construct are dependent in a large degree upon the character of the roads upon which they are operated. While it is true that the invention and development of the automobile has increased the demand for improved roads, it is also true that the growth of improved roads has increased the demand for and use of the motor vehicle, and future limitation in road construction will act as a limitation on the vehicle market.

It appears to me to be particularly unfortunate that there should be any controversy between the railroads and the users and builders of motor vehicles, instead of complete harmony and cooperation. Except in the most isolated cases, competition between these two forms of transportation is most unlikely. I think this is almost universally true with regard to transportation of goods; and in the transportation of passengers it is almost equally true if we stretch our imagination to embrace what must be the development of the future. I recognize the fact that there is a large amount of capital invested in street-railway transportation, but I am also impressed more and more daily with the fact that the streets of our cities are becoming less able to accommodate the burden of traffic that they are called upon to bear. It seems to me not at all improbable that this condition will make it imperative in the not very distant future to replace track vehicles with a more flexible form of vehicle for short hauls and where frequent stops are necessary.

This problem of highway capacity as evidenced by our city streets deserves the most careful study on the part of every automotive engineer, particularly as to what its probable effect will be on future design requirements as affecting the size of the vehicle, the control with respect to steering, turning-radius, acceleration and braking. In many of our cities very stringent regulations with regard to parking have been put into force. It is useless to spend our time in railing against these provisions, for in some measure at least they represent the legitimate effort to distribute the use of the streets in a fair way among all citizens. The problem presented is of the most complex nature and deserves careful study and analysis.

Another result of the increasing traffic-density is the lowering of the efficiency of motor vehicles as a means of saving time. As the cost of operation of motor vehicles has been reduced, and the possibility of use thereby increased, this new factor of limitation of speed, due to congestion, becomes increasingly important.

In the broader aspect of transportation in rural and suburban communities there should be practically no question of conflict between the railroad and the motor vehicle. We have in this Country a sufficiently close-up picture of the development of transportation facilities to be able to get a very comprehensive and intelligent view of the relation between various means of transportation and the establishment and development of communities.

The early settlements were along the seaboard and the more navigable streams, and this condition of affairs continued up to the time of the development of the railroad, which resulted in the unlocking of the vast inland empire and the linking-up of the Pacific coast with the Atlantic, which would have been practically impossible without this new means of transportation. The development of electricity and its application to high speed interurban lines was the next step in bringing high-speed transportation into closer contact with the small community and individual. It is obvious, however, that the operation of rail lines calls for a virtual monopoly of territory in the form of a franchise, and limits the operation of vehicles over any given track to one centralized authority, and calls for fixed schedules of operation.

The advent of the automobile has resulted in placing into the hands of the individual a smaller and more flexible unit with practically the equivalent speed-capacity of the railroad. This vehicle, capable of being operated over the road, can be made more truly competitive and infinitely more flexible and independent of fixed schedules. The growing use of the automobile and the truck, coincident with the development of and as an auxiliary to the railway system, has resulted in extensive suburban and rural development which would probably have been as impossible without the automobile as the development of the inland cities of this Country would have been without the railroad.

While this development has resulted, and the increase in realty value is recognized and acknowledged, the in-

*(Concluded on page 26)*

# Principles of Motorbus Design and Operation

*By* G. A. GREEN[1]

SEMI-ANNUAL MEETING PAPER *Illustrated with* PHOTOGRAPHS AND DRAWINGS

IN the paper an attempt is made to answer the broader phases of the questions: What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? by establishing the principles on which the design and operation of motorbuses should be based. The treatment of the subject is in the main impersonal, although specific references to the practice of the Fifth Avenue Coach Co. and illustrations of its equipment are made to emphasize the points brought out. The questions of the unwisdom of overloading, rates of fare and the service requirements are discussed briefly as a preface to the paper proper.

The factors controlling bus design are stated to be (*a*) safety, (*b*) comfort and convenience of the public and (*c*) minimum operating cost. The various subdivisions of each are commented on in some detail, and numerous illustrations and tabular data supplement the text. The conclusions reached are that trucks or automobiles, either modified or unmodified, are absolutely incapable of rendering satisfactory and economical service as buses; such failures of buses as have occurred were due to the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience; and, if the Society would concentrate its standardization work on the motorbus, much good could be accomplished.

THE questions that builders and intending operators are asking today are, What constitutes a bus? and In what respects does a bus differ from other classes of automotive equipment? There seems to be a general agreement that a properly designed bus has special requirements; that it differs materially from equipment such as trucks and automobiles.

I have been requested to give the Fifth Avenue Coach Co.'s views on this subject. It is, of course, possible to deal with only the broader phases. No attempt will be made to discuss detail design, but merely to establish the principles on which it is thought such design should be based. We believe that with problems of this character, it is principles that really count, that once having clearly established them, the rest is comparatively easy. Actually, there is no real mystery in motorbus design. It is purely an engineering problem and there is available ample engineering talent to afford its solution, but the principles must first be established.

In the preparation of this paper the underlying thought has been to treat the subject in an impersonal manner. Illustrations and specific reference have been made to our practices only when this has appeared to be the simplest and most direct method of approach.

### THE UNWISDOM OF OVERLOADING

We believe this question is of paramount importance, not only to the automotive industry but to all who are contemplating bus operation in any form. Our policy is predicated on a seat for every passenger. At the inception of our business this was our slogan. We have never departed from it and we never expect to do so. We are convinced that this policy has been, perhaps more than anything else, a factor in the building up of our enterprise.

It is, of course, possible to carry a certain percentage of standees in a vehicle, the spring-suspension of which has been correctly designed to carry properly a seated load. In our judgment, however, this figure should not exceed 30 per cent. But even this is unsatisfactory, for once standees are permitted, their limitation is most difficult.

Obviously, the problems requiring solution from the standpoint of spring-suspension are much less numerous with vehicles operating on rails than is the case with rubber-tired equipment running over roads. With the former, overloading has no immediate serious consequences—at least from the standpoint of the rolling stock. The spring-suspension with a bus must of necessity be a compromise between minimum and maximum loads. If the range is too wide, bad riding conditions must obtain during by far the greater percentage of the total time, for the packed loads will, generally speaking, occur only during the rush periods. This means that 90 per cent of the time there will be a state of discomfort. This will have an extremely bad effect on both the vehicle and its occupants. Another vital point to consider is that a bus is not kept in a comparatively straight and rigid course by steel rails. The advantageous flexibility of a bus in steering its course at will has its disadvantages if standees are permitted, for the shifting of the weight of the standees when the bus swerves tends to make it unsafe, throwing the passengers about inside the vehicle and rendering the operator liable to heavy damage and accident suits.

We are unqualifiedly behind any movement that will aid the bus to come into and remain in the field that is peculiarly its own. We are positive that the short road is the seated load and if builders will bear this in mind from the standpoint of design and warranty, the automotive industry will assuredly find ample repayment.

We earnestly hope that the automotive industry will read the writing that is so plain to see and that it will profit by what has occurred with the street railways, in regard to the matter of overloading. For it must be remembered that the bus has its limitations and that it is not the cure-all for every ill that transportation is heir to.

### THE MATTER OF FARES

Strictly speaking, there is no actual relationship between the design of a bus and the fares charged to passengers. Obviously, however, the better the design, the lower will be the operating cost. Naturally, this will make for lower fares. We believe that in the present state of the art no real success can be attained with less

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than a 10-cent fare. We are, of course, assuming operation based on seated loads and ample service during both the light and the heavy hours. But with character service, properly designed and maintained equipment, the people are quite willing to pay a 10-cent fare. There is ample evidence of this in New York City, Detroit, Chicago, Toronto, and other cities.

The necessity for a 10-cent fare does not rest with only the bus. Many electric railways need a 10-cent fare in order to be put on a paying basis. The last available tabulation shows that 140 electric railways in the United States are receiving a 10-cent fare, and that over 95 per cent of the electric railways in the cities of the United States have received varying increases in fare during the last few years. Some cities have a first fare of only 6 or 7 cents, but to this must be added a charge for transfers. Many cities have been placed on the zone system that works out in some cases as high as $3\frac{1}{2}$ cents per mile. Even with an increased fare, the last available figures show that about 10 per cent of the electric railways in the United States are in the hands of receivers.

It is not the purpose of this paper to enter into a lengthy discussion of operating costs, for unless this matter is treated in considerable detail, accurate deductions are almost impossible. Obviously, a correct comparison of operating expenditures can be made only on the assumption that similar detail classifications are employed in conjunction with a similar accounting system. Here the difficulties begin, for as yet few companies operating buses use the same accounting methods.

No doubt there are many who, while not desirous of making a minute survey of details of operating costs, would be interested in knowing something about this rather complicated matter other than mere expressions of opinion. For this reason there is shown in Table 1 not the customary detail cost statement, but what might be described as an income analysis. Actually it represents a distribution of the dime as received from each of those who rode on our buses during the year 1921.

TABLE 1—DISTRIBUTION OF EACH FARE RECEIVED

| | Cents |
|---|---|
| Total Operating Expenses | 6.50 |
| Total Taxes | 1.16 |
| Reserved for Injury and Damage Claims | 0.17 |
| Reserved for Depreciation | 0.29 |
| Interest on Capital Investment | 0.39 |
| Net Income | 1.49 |
| Total | 10.00 |

From these figures it is abundantly clear that we should have made a very bad showing with a fare of less than 10 cents. Here is emphasized very clearly the fact that the success of failure from the standpoint of an undertaking such as our own depends absolutely on the addition or subtraction of what at first sight appear to be insignificant amounts. To emphasize this point, during 1921 we carried a total of 52,216,946 passengers, so the net income from this source at 1.49 cents per passenger works out at $778,032.50. To permit of a comparison being made between the conditions confronting us and those faced by others, it should be noted that we operate a total of 25 miles of one-way route, that our longest run is 10.2 miles and our average haul 5.0 miles.

## The Bus and Its Service Requirements

Before discussing the bus from a design standpoint, something may be gained by outlining the character of service that must be expected, for it is here that the average engineer underestimates the difficulties to be encountered. First, let us consider the cumulative result of a year's performance of the physical limitations that are primarily responsible for wear-and-tear. For the sake of argument it may be assumed that these data are applicable to any bus operated by any public utility. The figures are presented in Table 2.

TABLE 2—DATA ON BUS OPERATION IN NEW YORK CITY

| | |
|---|---|
| Yearly Mileage | 30,000 to 60,000 |
| Stops and Starts | 180,000 to 360,000 |
| Change-Speed Applications | 360,000 to 720,000 |
| Clutch Applications | 360,000 to 720,000 |
| Different Drivers | 1,095 to 2,190 |
| Brake Applications | 200,000 to 400,000 |

Assuming the same general plan of upkeep as employed by the Fifth Avenue Coach Co., each bus would be thoroughly inspected after every 2000 miles of operation and rebuilt and repainted yearly. A vehicle would be expected to require no incidental repairs between inspectional periods and no major repairs between either inspections or yearly overhauls. The inspectional periods would occur approximately every 14 days. The maximum inspectional allowance is 8 hr. The allowance for yearly overhaul is 7 days. Roughly, it may be said that under these conditions, each bus is scheduled for service 358 days out of 365.

The statistics quoted as to mileage, stops and starts, and the like, speak for themselves. Those who have never had control of a public utility operating buses cannot possibly picture the sum total of the abuse the average bus must suffer. More than anything else, frequent changes in drivers result in increased service difficulties. It may be safely said that if one could with a bus have the same driver daily, at least 50 per cent of the service troubles would disappear. This, however, is quite impractical, since the loss in earnings would many times offset the decreased service cost. Even with an operation of moderate size, the bus must of necessity lose its identity. It becomes merely a transportation unit. There must be changes in drivers daily, many of whom will feel scarcely any pride of ownership. All they are concerned with is being on schedule time. This means that the bus will be subject to extraordinary abuse. The mechanisms of the bus must be capable of treatment of the most brutal nature; otherwise constant failures will occur.

Before one can proceed very far from a design standpoint, there must be some fairly clear conception of the vehicle life that is to be expected. In this connection it is necessary to lay stress on the fact that motorbus design is still in its initial stages. Five to 7 years is about the maximum life of the most modern type. It is not a matter of wear-and-tear, for a vehicle may be so well cared for that there is no limit to its life. Obsolescence is the real issue. The ideal conception is to carry out the design so that the various units which when assembled comprise the complete structure, have as nearly as possible an equal life.

## Controlling Design Factors

In its broadest sense we believe the controlling design factors from the standpoint of the motorbus, in the order of their importance, are

(1) Safety
(2) Comfort and convenience of the public
(3) Minimum operating cost

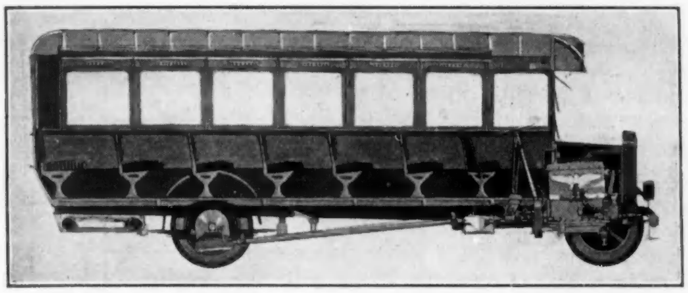


Fig. 1—Sectional View of the Type-J Bus

Safety easily heads the list and a very large proportion of the engineering development work must be concentrated under this heading. It is generally agreed that a truck carrying freight should be in all respects safe, and that every reasonable precaution should be taken to render automobiles transporting from 1 to 7 passengers safe; so how much more important is it that a vehicle carrying 50 or more passengers should be free from every sort of hazard! It must be remembered that much of the mileage of the bus is through congested thoroughfares. This is not the case with the average automobile or truck. Again, the average individual makes some effort to get out of the way of a truck or automobile, but the bus, with its acknowledged flexibility, is supposed to move out of the paths of both vehicles and pedestrians.

The design of a motorbus from a safety standpoint includes certain basic features which must be incorporated in the general constructional plan. There are also other detail features which must be included. The latter are dictated by human considerations. Reference is now being made to providing the driver with reasonable comfort and convenience so that no undue hardship will be inflicted upon him as a result of the performance of his duties. First, let us consider the former. These are

(1) Low center of gravity
(2) Wide frame, track and spring centers and general dimensions
(3) Effective brakes
(4) Short turning-radius

### Low Center of Gravity

Beyond doubt, the future bus will be low hung. The inherent danger in connection with any other form of

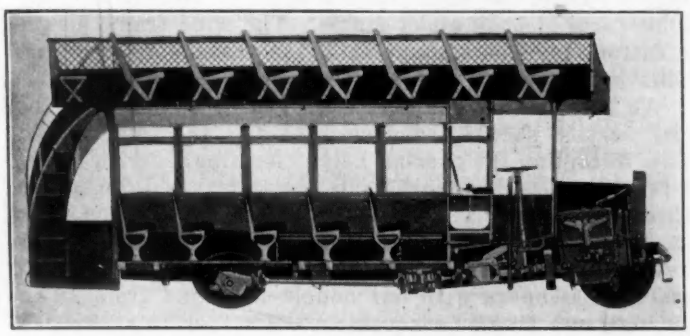


Fig. 2—Sectional View of the Type-L Bus

construction is the possibility of overturning. Under conditions of proper operation, the hazard may be non-existent, but we have always before us the possibility of human failure. Actually the danger is much more real than apparent. The controlling element governing overturning is centrifugal force. Vehicles seldom if ever overturn as a result of high speed and sudden impacts or brake applications. Overturns are almost invariably due to a combination of speed and turning-radius. The only reliable guarantee against this class of accident is a low center of gravity.

In many cities there are overhead wires and various other obstructions. The low bus is often a necessity to pass under such obstructions. Certainly, the lower the vehicle, the less the hazard. These remarks apply particularly to double-deck vehicles. With the single-deck vehicle, the higher speed is a factor that must be fully taken into account. Entirely apart from the matter of safety, a low-hung vehicle has a more graceful appearance. There is less time lost in boarding and alighting, there are fewer boarding and alighting accidents, and the schedule speed can be faster. Lastly, assuming proper

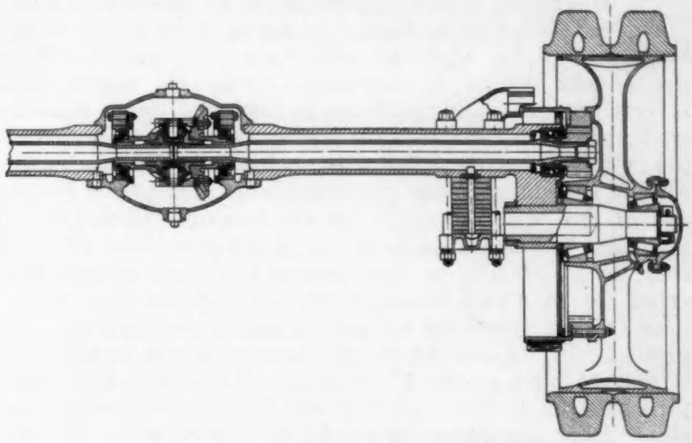


Fig. 3—Sectional Drawing of the Type-L Axle

design, a low center of gravity results in improved riding properties.

We have found that a safe and practical height of the frame from the ground for a single-deck bus is 25 in. and for double deck bus, 18 in. The center of gravity of our type-L double-deck vehicles, with a full complement of passengers on both decks is 52 in. from the ground. With our type-J single-deck bus, this dimension is 38 in. It is interesting to note that when rounding corners, even at a high rate of speed, skidding will occur due to centrifugal force and overturning is scarcely possible. Furthermore, rolling or sidesway is practically eliminated. The sectional views of our J and L-type buses reproduced in Figs. 1 and 2 indicate clearly how this condition has been reached. With type L it will be seen that the frame and rear-axle construction is somewhat unconventional. The rear axle is of the internal-gear type. The spiral bevel-gear and differential assembly is in unit form and can be entirely assembled and adjusted on the bench. The carrying member is a heat-treated forged job.

From the sectional drawing shown in Fig. 3 the general construction of the type-L axle will be clear. It

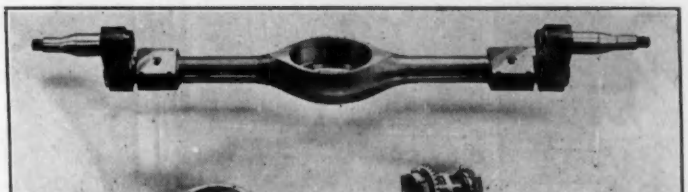


Fig. 4—The Type-L Rear Axle

will be seen that the ends of the carrying member are cranked, the wheel spindles being above the drive-shaft center-line. It is in this manner that the low-level feature has been accomplished. The photograph showing the carrying member and driving-gear assembly which is reproduced in Fig. 4 at once emphasizes the general simplicity and accessibility of construction. Due to the fact that the drive-shaft pinions are in the vertical plane, a special form of tooth has been developed for the internal gear to provide adequate clearance and at the same time permit of maximum silence even after a certain amount of wear has occurred.

We do not employ this special form of axle construction for the type-J bus. This class of vehicle will have a much wider use; therefore, the matter of road clearances must be taken into account. In many cases single-deck vehicles will be operated over very bad roads. The double-deck vehicle is essentially a city job where the streets are, generally speaking, in fair condition. Again, with the single-deck vehicle, the floor-level requirements are not so exacting. There is no top deck to take care of, and the entrance can therefore be located at the front end of the bus; but with the double-deck vehicle, conventional practice is to have the passengers enter at the rear, so in passing to the interior they are obliged to cross the rear axle which must be of special design to have the floor level within easy stepping distance of the ground. In the case of the single-deck bus it is not desirable to have a step 18 in. high. Therefore, the best plan appears to be to employ an orthodox rear-axle design. Even assuming the use of our type-L rear axle, it would not be practical to produce a stepless vehicle. The

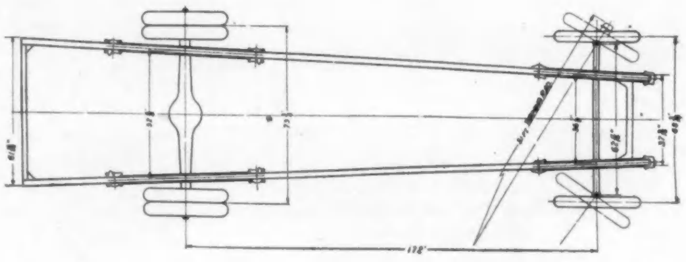


FIG. 6—CHASSIS DIMENSIONS OF THE 29-PASSENGER SINGLE-DECK BUS

appearance would be completely spoiled and, as explained above, the ground clearance would be cut to a point where the vehicle would be unsuitable for use in many localities. Of course, a stepless single-deck vehicle can be produced, but its practical value for general utility purposes is debatable.

Among the constructional difficulties in connection with the production of low-level equipment, one of the problems is to obtain a flat floor. There is a natural tendency for the components to project above the frame and therefore through the floor. To avoid this, special design is required. The effect of a flat floor is very pleasing to the eye. Its structural strength is greater. It is less costly to keep in repair and there is less possibility of accidents due to the passengers' feet coming into contact with the obstructions during the boarding and alighting processes. The view through the door of our type-J vehicle, Fig. 5, brings out this point to advantage.

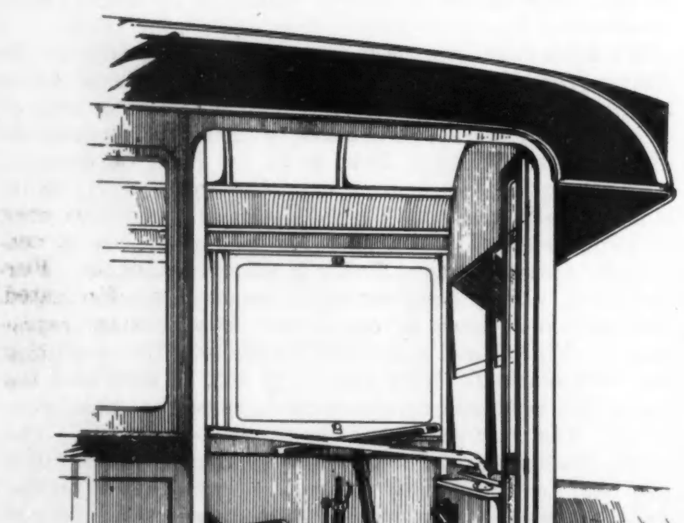


FIG. 5—VIEW THROUGH THE DOOR OF THE TYPE-J BUS

### WIDE FRAME, TRACK AND SPRING CENTERS

These features are necessary to provide for adequate vehicular stability and, in conjunction with a low center of gravity, make for maximum safety. The necessity of providing proper stability applies equally to single and double-deck vehicles. It may be said that the added risk due to the top-deck load with the latter is more than equalled by the faster speed of the single-deck unit.

Apart from the matter of safety, a wide frame is necessary in connection with the body construction. Obviously it is desirable to support the body as far out as possible, for in all cases the seating arrangement is such that the passengers are grouped about the outer edges. Then, the wide frame admits of the lightest possible form of body under-frame. The wide frame also is a factor from the standpoint of the passengers' comfort. This point will be referred to later.

We believe that the overall length of a motorbus for city service should not exceed 26 ft.; the total width, 7 ft. 6 in.; and the over-all height for single-deck vehicle, 9 ft. With the double-deck bus, the last-named dimension should be such that a person standing on the top deck can clear a 14-ft. structure. With these dimensions we have found it possible to accommodate comfortably 51 seated passengers with our double-deck, and from 25 to 29 with our single-deck vehicle. Whether this practice is economically correct for all localities, we cannot say. We have, however, up to the present found that this arrangement works out very well both in our own service and in the service of those who have purchased our equipment.

Next, there is the question of important dimensions other than those over-all, such as the wheelbase which naturally affects the axle load distribution, the turning-radius and the general comfort and balance of the vehicle. For the class of vehicle now under discussion, we believe

that this dimension should not be less than 168 nor more than 180 in.

The front track should be ample in width and not less than 67 in., for to turn a bus within the intersection of the average city street, it is necessary to move the front wheels through an angle of not less than 35 deg. This determines the distance between the front-axle pivots and the springs. The spacing of the front springs should not be less than 36 in., since they are responsible to a large extent for the stabilization of the vehicle when turning a corner.

Regarding the rear track, we believe that the outer edge of the tires should closely correspond to the extreme over-all width of the body and that the rear springs should be as close to the tires as is practical. For buses as above described, the rear track should not be less than 72 in. This will bring the distance between the springs to approximately 52 in. Having decided the approximate distance between the vehicle springs, it naturally follows that the best design is to arrange the frame dimensions so that they connect with the springs in the closest and most practical manner. Our practices in regard to these matters may be readily followed from the diagrammatic sketch of the type-J chassis as shown in Fig. 6.

### Effective Brakes

Perhaps the most difficult problem that engineers must face is the brake question. Even now it has not as yet been solved entirely satisfactorily, at least insofar as our knowledge goes. With the bus, the number of applications is in excess of that of the average truck or automobile, and the brakes of a bus must be sufficiently powerful to lock the wheels at any moment. Yet the effort required for average application must not be such that a driver may become exhausted as a result of the work imposed upon him.

Particular attention must be paid to the location of hand-brake lever. It should be positioned so that it can be grasped firmly without moving the body out of the normal seated state. We believe the best practice is to have the lever arranged for a push and not a pull-on. Time can thus be saved, and a fraction of a second is often the determining factor from an accident-prevention standpoint.

The brakes of a bus must be free from undue noises such as squeals or rattles. This means, among other matters, the use of special brake-drum material. The conventional soft pressed steel is practically useless. The best plan is to employ treated steel forgings or, failing in this, steel castings with a high carbon-content.

The friction surfaces must have long life, and the adjustment be such that no tools or special skill are necessary. We attach considerable importance to the matter of foolproof adjustment. The J system as illustrated in Fig. 7 shows our method. It will be seen that there are two vise-like levers. The outside controls the hand, the inside the foot brake. One turn is usually sufficient. If by any chance the levers are not returned to the vertical, they will automatically reach this position by force of gravity.

The braking action must not be too abrupt. It must be positive yet not sudden and violent, for such a condition is exceedingly severe on the driving members, tires and body. It is also a frequent source of accidents from which serious claims may result. Brakes must be sufficiently good, yet not too good. Excessively efficient brakes have a most marked influence on tire wear. It may be said that tire wear is almost directly proportionate to the effectiveness of the brakes.

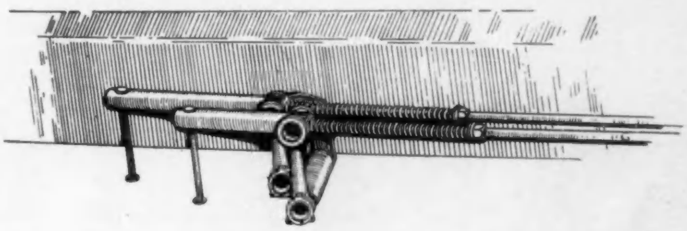


Fig. 7—Outside Brake Adjustment

In bus operation it is desirable from every point of view to cover the route as quickly as safety will permit. In this manner the maximum number of passengers can be carried daily. With a fixed maximum-speed, this means fast deceleration and acceleration. Expressed in another way, the problem is to move from a stop in one location to a stop in another in the least time. In our own service this must be done without exceeding a speed of 15 m.p.h., or accelerating or decelerating faster than 2 m.p.h. per sec. A still more rapid rate of deceleration is, of course, available for emergency, but it will be uncomfortable and unsafe, especially for standees.

The acceleration and deceleration graph as reproduced in Fig. 8 shows how closely the present type of equipment approaches this conception. To make the test, one of our double-deck buses was selected at random.

### Short Turning-Radius

One of the great advantages of a bus over any other form of transportation unit is its flexibility. A bus can be switched around at any point, and it is highly desirable that it should be able to make a complete turn in the average thoroughfare without backing, for the latter practice if followed in congested areas merely adds to both confusion and congestion. There is also a marked possibility of an increased number of accidents.

A short turning-radius is dependent on the interference of the tires with the drag-link, front springs or frame,

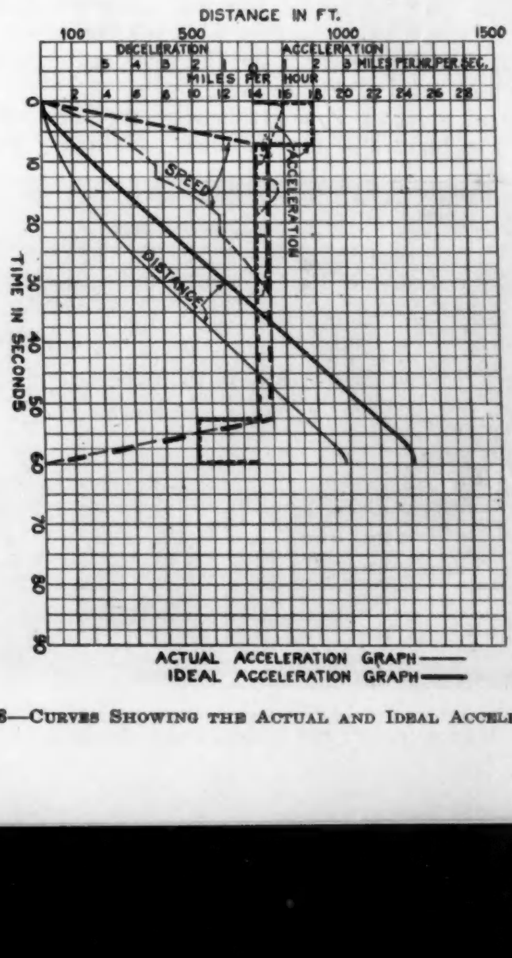
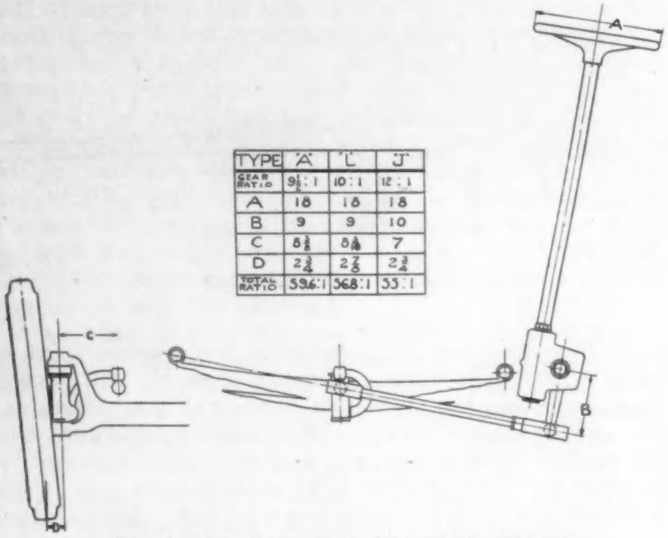



Fig. 8—Curves Showing the Actual and Ideal Acceleration




| TYPE | "A" | "L" | "J" |
|---|---|---|---|
| GEAR RATIO | $9\frac{1}{2}$:1 | 10:1 | 12:1 |
| A | 18 | 18 | 18 |
| B | 9 | 9 | 10 |
| C | $8\frac{3}{8}$ | $8\frac{1}{16}$ | 7 |
| D | $2\frac{3}{4}$ | $2\frac{7}{8}$ | $2\frac{3}{4}$ |
| TOTAL RATIO | 59.6:1 | 56.8:1 | 55:1 |

FIG. 9—DIAGRAM OF THE STEERING LEVERAGES

when the wheels are turned at the maximum angle. The controlling elements are wheel-spring tracks and wheelbase. As the radius of the steering angle equals the wheelbase divided by the sine of the front-wheel lock, it can be seen that a wheelbase of reasonable length is important to secure a short turning-radius.

From the viewpoint of safety, the design features dictated by human considerations are

(1) Easy steering
(2) Clear vision for driver
(3) Comfort and convenience for driver

### EASY STEERING

The steering of a bus should be at least as easy as that of the average automobile. To operate a stiff steering-gear is a hardship that certainly should not be inflicted upon the driver of a public-service vehicle. A driver's energy and effort must be concentrated on his regular duties, and if he becomes fatigued through the expenditure of unnecessary effort, faulty operation is bound to result. This means possible accidents. Tests have convinced us that the actual physical labor imposed on the driver of a bus in connection with the manipulation of a steering-wheel represents by far the greater proportion of the sum total of his work.

Ease of steering is controlled by the total ratios between the hand and road wheels. Naturally frictional losses in the steering-gear box and steering-knuckles are of importance. Minimum losses in these respects are dependent upon the use of properly lubricated anti-friction bearings. Another very important matter is that the steering-knuckle pins should lie in the vertical plane; otherwise there will always be a tendency to lift the front end of the bus when turning the steering-wheel. An angle in either the longitudinal or transverse plane will cause lifting at the expense of effort on the part of the driver.

It is highly desirable that there should be an absence of shocks at the steering-wheel. This is largely controlled by the total ratio, but also by the distance between the point of contact of the wheel and the road and the intersection of the knuckle center-line and the road. Every effort should be made to keep this distance small. With the J type the length of the lever arm is about $2\frac{3}{4}$ in.; and an increase of only 1 in. would decrease the total ratio some 36 per cent. This is the only point in the steering linkage where a change increasing the total reduction does not result in increased steering-wheel travel for a given lock. A short drag-link or the incorrect alignment of the drag-link with the front springs will also result in shocks at the steering-wheel when passing over rough roads.

Minimum steering-wheel travel is important as it makes a change of hand position unnecessary for ordinary driving. It also decreases the apparent back-lash, which is present in all steering mechanisms. The steering-wheel travel is roughly inversely proportional to the total ratio, which is kept as low as possible for this reason. Our practice so far as the important dimensions referred to above are concerned may readily be followed from an examination of the diagram of steering leverages as illustrated in Fig. 9.

### CLEAR VISION FOR DRIVER

This very important feature can be accomplished only as a result of joint chassis and body design. The driver should be located close to the left-hand side. This permits him to observe and also to signal his intentions to oncoming traffic. There should be absolutely nothing obstructing his view. He should face clear glass. It should also be mentioned that with single-deck vehicles the placing of the driver well over on the left-hand side provides for the very necessary boarding and alighting space for passengers and adequate room for operation of door.

Briefly, a driver's vision should be such that when seated, even back of a closed windshield, he will have nothing on which he can readily concentrate, no vertical posts or obstructions of any kind. He should just naturally sense that he is in the open. The illustrations of the front end of our type-J bus reproduced in Figs. 10 and 11 bring out this point with marked clearness.

### COMFORT AND CONVENIENCE FOR DRIVER

This is largely a question of seat formation in conjunction with the correct positions for brake, change-speed levers, pedals, accelerator, etc. Obviously, it is not a practical matter to give the driver of a bus as much room as with a touring car; therefore, much care and thought must be paid to the placement of pedals and levers. The conventional cowl as used in automobile practice is almost out of the question, for anything that tends to increase the over-all length of the vehicle is distinctly undesirable, particularly if such increases add nothing to the passengers' seat or pay-load space.


FIG. 10—A TYPE-J 25-PASSENGER SINGLE-DECK BUS

The driver should be comfortably seated at all times.

He should be able to reach his change-speed or brake levers without body movement. He should have ample leg-room and not be obliged to cramp his limbs when his feet are either on or off the pedals. To some extent this point is brought out in Fig. 5. The value of the flat floor from the standpoints of both passengers and driver, is apparent; also the side control without which there is of necessity a considerable loss of most valuable space.

### Comfort and Convenience of the Public

The American public is automotively inclined and the percentage of those owning cars is so large that when riding in any self-propelled vehicle, there is a natural tendency to compare its behavior with that of an automobile. In designing a bus this factor must under no circumstances be lost sight of. The success of any public utility depends on the good will of the public. It has been correctly stated that the permanence of any business depends upon the good will of those it serves and

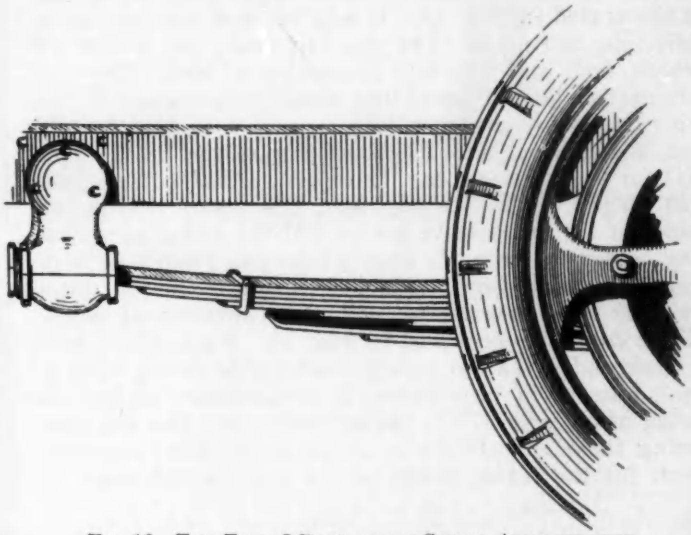


Fig. 12—The Type-J Progressive Spring Arrangement

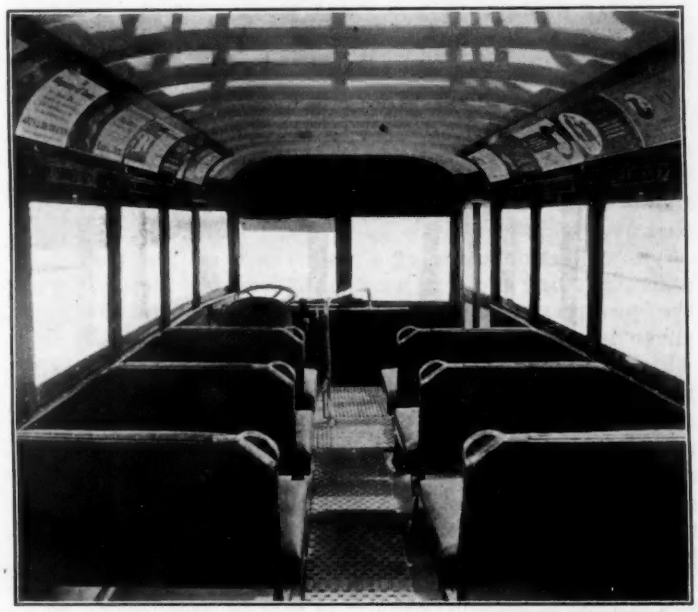


Fig. 11—View from the Rear End of a Type-J Bus

that no business can achieve permanent success that does not give in exchange for its earnings at least an even measure of helpful service. This applies especially to public utilities, and the truth has been abundantly proved in connection with the operation of our enterprise.

From the viewpoint of design, it is essential that consideration be paid to the attitude of the public as a whole. It is not enough to consider only the attitude of the actual riders; regarding the matter of comfort from these somewhat different angles, it is necessary that attention be given to

(1) Riding ability
(2) Reliability
(3) Silence of operation
(4) Smoothness of starting and stopping

### Riding Ability

Broadly, this is a matter of proper spring-design. There are, however, other important influences; the wide frame, track and spring-centers bear materially upon this question, for the nearer the wheels are to the outer edge of the body, the less will be the movement to which passengers must be subjected when obstacles are passed over. Again, with the wider track, many of the ruts and depressions created by vehicles of narrower gage, will be passed by. Incidentally, this is quite an important matter from the standpoint of road wear. The wide track also diminishes the wheel-pocket projection inside of body. The modern tendency is to employ cross seats and with the narrow-gage vehicle the wheel pockets are a source of much discomfort to those seated upon the inside immediately over them. A rigid frame, correct axle-load distribution and minimum overhang are all factors that make for better riding performance.

Apart from the points briefly touched upon above, the controlling factor from the standpoint of riding ability is, of course, the design of the suspension itself. Obviously, the difficulty is to obtain good riding under all conditions of load. Spring design is always a compromise; a spring must be able to withstand maximum load, yet vehicles are expected to ride reasonably well when light. As a matter of fact, they seldom, if ever, do so. In general, more damage is done to vehicles when running light than heavy because the riding properties under these circumstances are at their worst and the speed too often is high. Under conditions of heavy load, springs function best, and at the same time there is less likelihood of excess speed.

We believe that the answer will be found largely in

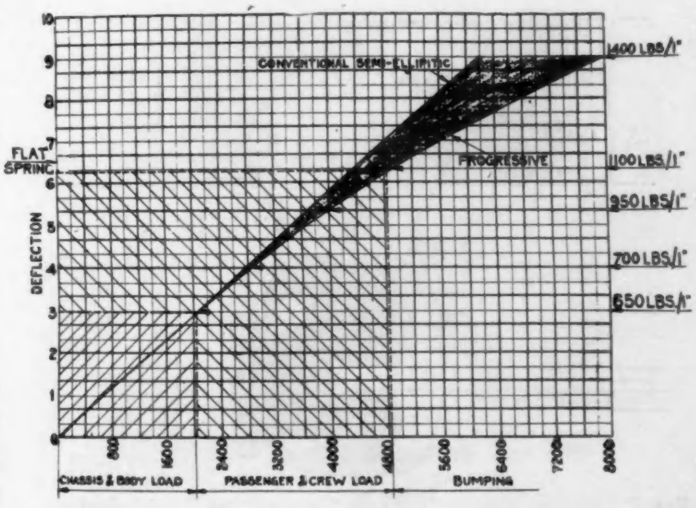
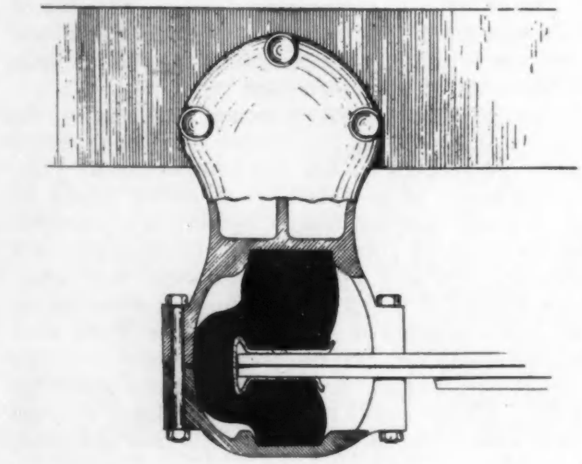



Fig. 13—Curve Showing the Comparative Deflections of the Progressive and Conventional Semi-Elliptic Rear Springs

the employment of what we term the progressive spring as illustrated in Fig. 12. It will be seen that spring is split into two parts. The top half takes the weight of vehicle, body and a certain proportion of load. The bottom part or helper, comes into action progressively. The top part must make a rolling contact with the bottom. One of the great advantages of this system is the fact that for no additional cost or weight, a marked improvement in performance is possible. The theory behind our choice of the progressive spring and the advantages that may be derived from its employment can readily be seen from an examination of the rear-spring deflection curve for both the progressive and the conventional semi-elliptic designs reproduced in Fig. 13. No doubt it will be appreciated that to secure comfortable riding with a small number of passengers, it is necessary to have a spring of not over 670-lb. per in. deflection. But a spring having these characteristics is not a practical arrangement, for the result would be too great a difference in

TABLE 3—DEFLECTION FOR PASSENGER LOAD

| | Conventional Semi-Elliptic Spring | Progressive Spring |
|---|---|---|
| Full Passenger-Load | 4¼ | 3¼ |
| Maximum Bumping-Load | 8½ | 6¼ |



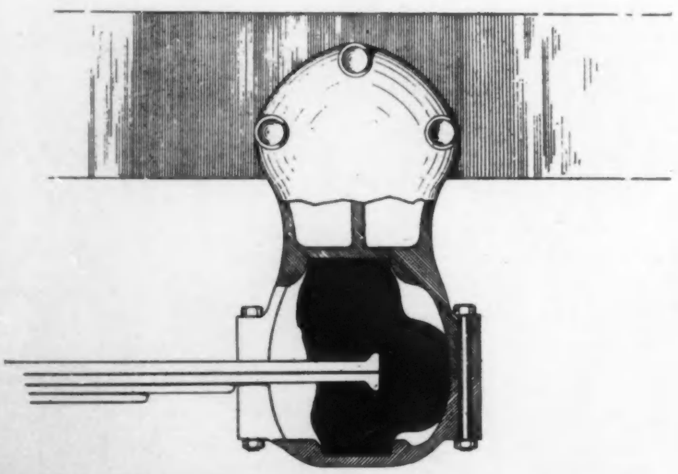


FIG 14—RUBBER SPRING-SHACKLES USED ON THE TYPE-J SPRING SUSPENSION

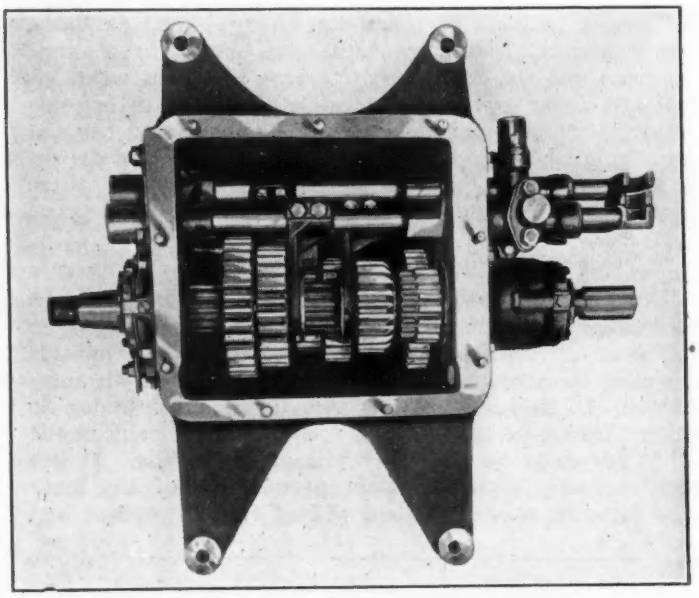


FIG. 15—THE FOUR-SPEED GEAR TRANSMISSION

body and step height between the minimum and maximum number of passengers. This point is clearly shown in the graph where the proportion of the 51-passenger load equals 2800 lb. per rear spring, from which the comparative figures given in Table 3 are deduced.

The deflection curve of a simple semi-elliptic spring is a straight line showing a constant load per inch. But as the progressive element comes into play gradually, a curve is apparent. The departure from a straight line which is shown shaded represents the load carried by the progressive element which can be designed to come into action at any desired point. It has been found most satisfactory to design this spring so that the stiffened action begins very gradually, that is to say, after a limited number of passengers have been taken on. Obviously, as the progressive element comes into action, there is a gain in the stability of the vehicle.

From the graph above referred to it is exceedingly interesting to note the change in rate of progression as a result of a variation in passenger load. The figures based on increments of 10 passengers given in Table 4 bring this point out in a striking manner.

TABLE 4—CHANGE IN RATE OF PROGRESSION FOR VARIATIONS IN LOAD

| No. of Passengers | Load per 1-In. Deflection, lb. | Increased Stiffness, per cent |
|---|---|---|
| 0 | 670 | 0.0 |
| 10 | 780 | 16.4 |
| 20 | 810 | 20.9 |
| 30 | 850 | 26.9 |
| 40 | 900 | 34.4 |
| 50 | 1,080 | 61.3 |

For our single-deck equipment we have standardized the Mack type of rubber shock-insulator which is illustrated in Fig. 14. This is by special arrangement with the Mack company. We are experimenting with this device for our double-deck vehicle but as yet are not prepared to state the results. This arrangement, in conjunction with our progressive system, markedly improves the riding conditions. It also avoids the necessity for lubrication and for replacement of shackles, shackle-pins and bushes; also, no spring-eyes are required. Experi-

ence up to the present shows that we may expect a very satisfactory life from rubber blocks.

### Silence of Operation

It is a problem to produce a silent vehicle. It is doubly a problem to retain this state throughout the life of the vehicle. Silence necessitates freedom from engine vibration, quiet transmission gears, evenly stepped gears, a quiet rear end, and generally the elimination of all rattles

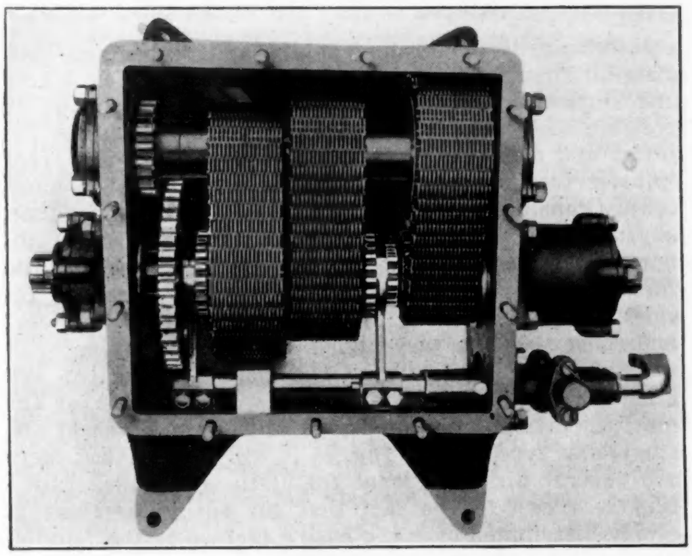


Fig. 16—The Three-Speed Chain Transmission

and squeaks from both body and chassis. To attain this, every detail of design must receive the most minute care. Silent operation is necessary in crowded thoroughfares, and certainly the people demand this condition in the residential areas, particularly at night when the streets are comparatively empty and noises become automatically emphasized. As a rule, noises are tolerated simply because such things are nearly always with us, but in the quiet of the evening sounds that ordinarily pass unnoticed become startlingly evident. In connection with the general question of noise it is interesting to consider

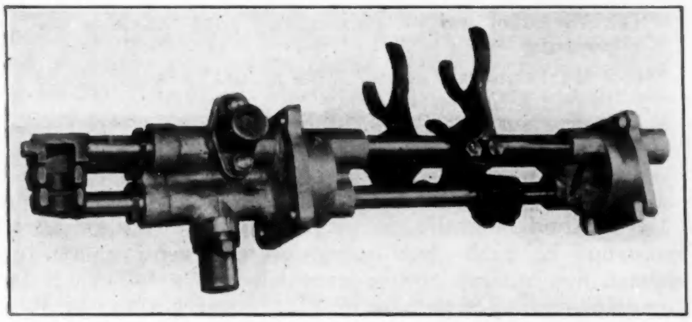


Fig. 17—The Shift-Rod Assembly of the Four-Speed Transmission

for a moment conditions on Fifth Avenue in the rush period during which we operate 180 buses per hr. in each direction. If this vehicular volume were not reasonably quiet, we should soon be ordered off the streets as a public nuisance and a menace to health.

From the standpoint of silence, our greatest difficulty has been and still is the matter of transmission gears. We employ a four-speed gear and three-speed chain transmission, shown in Figs. 15 and 16 respectively, de-

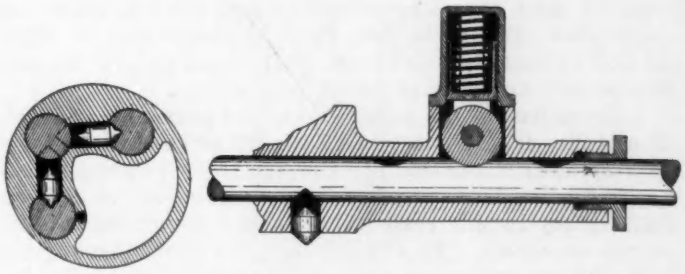


Fig. 18—The Locking Mechanism Used on the Shift Rod of the Four-Speed Gear Transmission

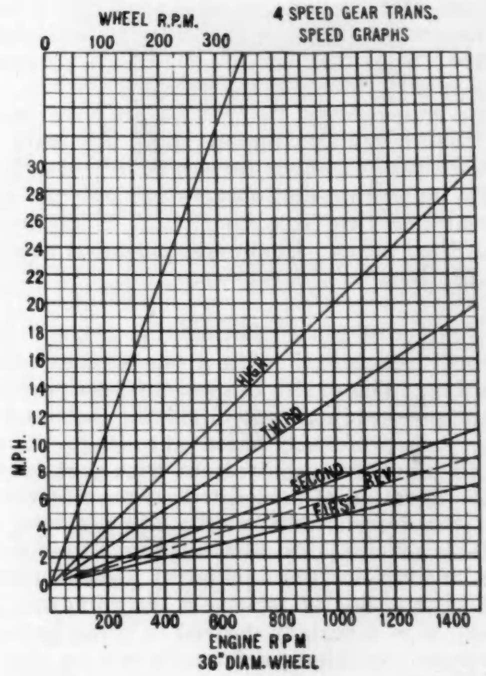



Fig. 19—Speed Curves of the Four-Speed Gear Transmission

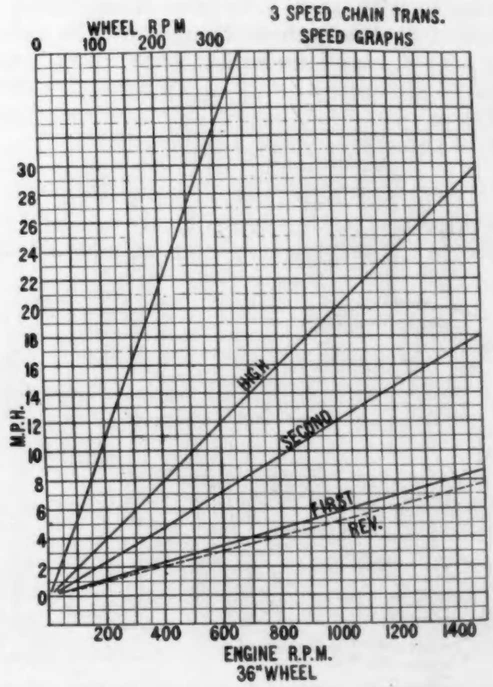



Fig. 20—Speed Curves of the Three-Speed Chain Transmission

pending upon the class of service and general operating conditions. It will be seen from an inspection of Figs. 17 and 18 that the shift-rods, their bearings and the lock mechanism are of substantial proportions.

Illustrations of the speed curves are presented in Figs. 19 and 20. It is worth while noting that the ratios of the four-speed transmission are almost exactly in geometrical progression. The three-speed transmission is not so satisfactory in this respect but here a compromise is of course necessary. This remark applies to all three-speed jobs. Where grades are severe, four speeds are highly desirable, to cut down ability losses to the minimum. But where roads are practically flat, the advantages of a four-speed transmission are not nearly so marked.

The silent-chain transmission is particularly useful for city service where there are frequent stops and starts, and where the percentage of direct-gear operation is relatively small. Substantially it is similar to a constant-mesh gear transmission but chains are used in place of gears. The shift is extremely short and very easy to effect. Such transmissions remain quiet throughout their useful life, and from our observation one can expect at least a year's service from the chains, which are cheaper to replace than gears. Chain transmissions are standard practice for London bus service.

### Reliability

The word "reliability" with a bus attains an entirely new meaning. The entire design must be predicated on ability to give uninterrupted service between clearly defined periods, preferably based on mileage. The ability of a bus to fulfill this requirement with particular reference to the duration of period will at once determine the utility of the design. The public will not long tolerate an unreliable service. Failures with an automobile cause confusion enough but the number of persons involved as compared with a bus is relatively insignificant.

One point it is especially desired to bring home is that under average conditions, drivers cannot be expected to make any attempt whatever to spare their equipment. All they are concerned with is stopping for passengers, avoiding accidents, and keeping in their places on the road in accordance with their schedule. Everything must be subordinated to these three things, and in cases where vehicles cannot stand up under such conditions, either the required changes must be made to enable them to do so or they should be scrapped, for assuredly they have no place in the operation of a public utility.

### Smoothness of Starting and Stopping

Smoothness of starting is primarily a clutch function, but of course the driver is a factor. Correct gear-ratios,

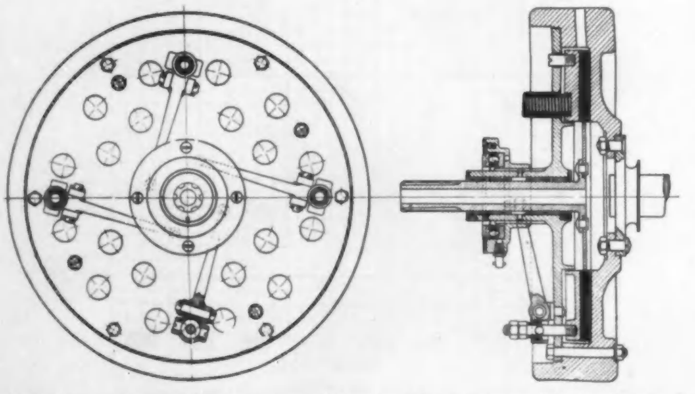


Fig. 21—Section Through the Clutch Used on the Types-J and L Buses

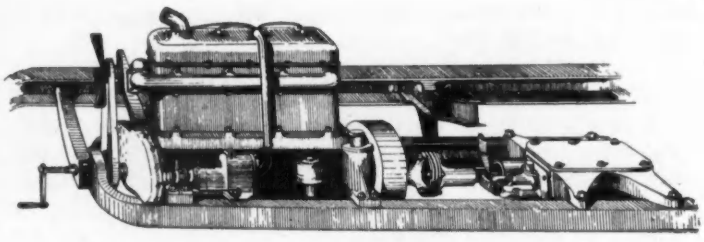


Fig. 22—Sub-Frame Mounting of the Type-J Bus

a satisfactorily performing engine and proper axle-load distribution are contributing influences. Quick starts and stops are highly dangerous from the viewpoint of possible accidents. Some of the heaviest claims for injuries and damages result in this manner. Apart from injuries to passengers, quick starts and stops do more toward causing damage to the chassis and the bodies than anything else. All driving members are subject to abnormal stresses with the former. With the latter, the fore-and-aft or lateral movement, which of necessity results, causes a loosening up of post joints, panelling, etc., and consequently a very high rate of depreciation.

Of the various features that make for efficient and economical operation, the clutch is perhaps one of the most important. We employ exclusively a clutch of the single-disc type. From Fig. 21 it will be seen that there are several unconventional features. Particular attention is drawn to the fact that the spring pressure is evenly distributed over the entire surface of the friction members by 20 small springs, the levers are balanced against centrifugal force and the disc is exceedingly light, thus simplifying the changing of gears. Incidentally, a clutch-stop has been found unnecessary. The removal of the clutch body is an extremely simple operation, as is also the adjustment of the levers. Smoothness of stopping is discussed under the heading of Brakes.

Minimum operating cost demands:

(1) Maximum accessibility
(2) Minimum consumption of labor and material. This of course means excellence of both materials and workmanship
(3) Minimum consumption of fuel
(4) Minimum weight, particularly that which is unsprung
(5) Maximum safe speed. This naturally comprehends rapid acceleration
(6) Maximum tire-mileage

### Maximum Accessibility

It is fundamentally necessary that the design of a motorbus be such that inspection and repairs can be carried out quickly and economically. We believe it is imperative that separate unitary construction be followed. For instance, engines, carbureters, all electrical equipment, fans, clutch couplings, transmissions, control levers, axles, wheels and propeller-shafts should all be entities unto themselves, so that the repair of any one of these assemblies will not necessitate the removal of any other.

As a practical illustration, take the orthodox unit powerplant and assume it is necessary to renew the clutch friction linings. The propeller-shaft, transmission and complete control system must first be taken down, possibly even the engine moved forward. In all probability the vehicle must lose a complete day's service. Compare

this for a moment with the relatively simple operation where the separate-unit form of construction is employed, such as with our J or L types. Here we need only remove a few bolts from the clutch coupling and housing. The clutch can then be taken out as a complete unit and the linings replaced within a period of 20 or 30 min. To picture this condition, there is illustrated in Fig. 22 our form of subframe mounting.

The unitary system, if properly carried out, guarantees minimum loss of bus-hours, minimum operating cost, and minimum difficulties from the standpoint of training employes. Obviously, less skill is required on the part of mechanics where they are constantly performing the same operation; here it is simply a question of specialization. But where the construction is such that multi-repair operations are required, the situation is much more complicated. Summing up, to be obliged to remove several units before a faulty unit can be inspected, repaired or replaced, is a condition not to be considered for a moment. Such practice would be ruinous from a public utility standpoint.

It must be remembered that the general conditions surrounding repair work are seldom ideal. There is the matter of wet floors, dirt surrounding the various units, often lack of light. Garage repair forces must work Saturdays and Sundays, which is not particularly attractive. In actual practice it is exceedingly difficult to find men who are willing to work nights. Taken as a whole, the conditions surrounding the work of the repair-men seldom bear favorable comparison with modern high-class factory practice. Here again we wish to emphasize the desirability of unit construction, for the theory is to remove the defective unit and take this to a central repair plant having all the advantages of the modern factory, so that the repairs can be promptly executed by skilled men working under the best possible surroundings.

In connection with the matter of accessibility, it should be remembered that repairs and adjustments must be occasionally carried out at night, sometimes under most unfavorable conditions. Again, assuming the use of low-level equipment, the design should be such that inspections, repairs and renewals can in practically all instances be undertaken from the sides or underneath the vehicles. This means the use of pits. The practice of providing trap-doors inside buses is not desirable. Trap-doors weaken the bodies, are a possible source of accidents, cannot be kept tight in place, permit exhaust gases to leak through, and create undue noise. Experience has shown that it is highly unsatisfactory to carry out chassis repairs from the inside of the body. If this practice is indulged in, claims are bound to result from passengers due to their clothes coming into contact with grease or dirt. Mechanics are sometimes careless and this results in unnecessary damage to the interior fittings, particularly the seat cushions.

### Minimum Consumption of Labor and Material

From a financial viewpoint, the success or failure of a utility operating buses depends upon the cumulative additions or subtractions of small amounts expended on either labor or material. Sometimes the items may appear insignificant but, taken as a whole and over lengthy periods, the story is entirely different. When working, a bus is a heavy consumer of both labor and material. The consumption is perhaps much greater than is generally supposed. To afford a practical illustration, Table 5 representing the actual consumption by our company of some of the major elements for the year 1921,

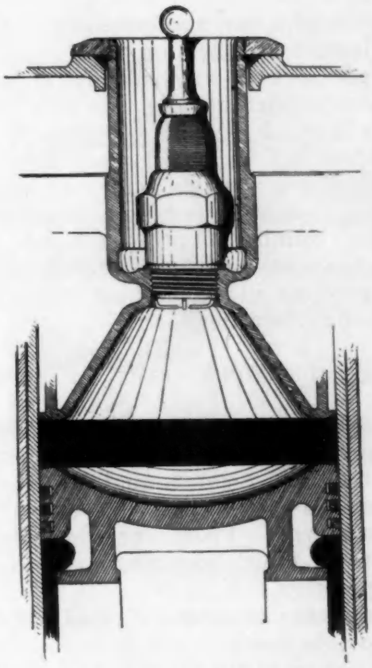


Fig. 23—Sectional Elevation Through the Combustion Chamber of the Engine Used on the Types-J and L Buses

may be of interest. These figures are based on the average of all buses.

TABLE 5—FIFTH AVENUE COACH CO.'S COST PER BUS FOR 1921

| | | | |
|---|---|---|---|
| Gasoline | | | $1,125.94 |
| Lubrication | | | 109.42 |
| Tires | | | 284.34 |
| Repairs to Chassis | Labor | $676.97 | 1,436.78 |
| | Material | 759.81 | |
| Repairs to Bodies | Labor | 359.00 | 521.44 |
| | Material | 162.44 | |
| Drivers | | | 3,071.71 |
| Conductors | | | 2,692.48 |
| Total | | | $9,242.11 |

From a casual study of these data it will be seen that a relatively small percentage of saving, if applied to any of the items and then multiplied by a large number of

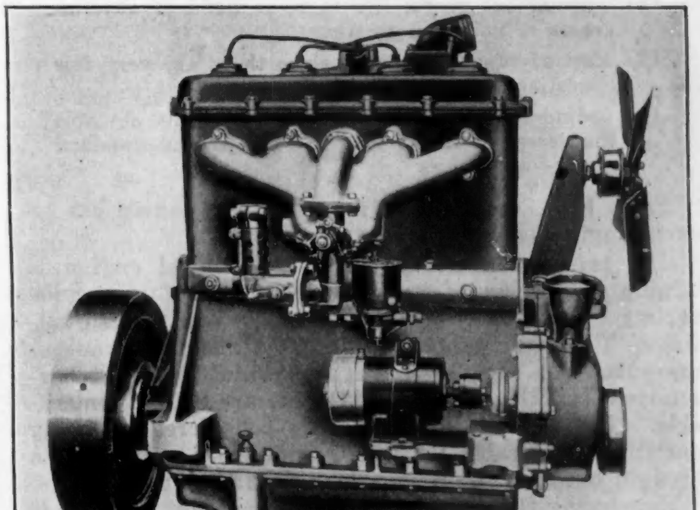


Fig. 24—The Engine Used on the Types-J and L Buses

vehicles, must total a vast sum annually. If one assumes that the equipment in question is of good design and that its maintenance is economically undertaken, then how much more important does this issue become when the reverse is true.

Perhaps it will not be out of place here to point out that the profit of the average utility expressed percentagewise, usually does not run beyond one figure, and that there are a vast number of utilities where the figure is in red. To change the color and to exceed the single-figure basis, requires all that is best in design, material, workmanship and operating care.

### Minimum Consumption of Fuel

Aside from the human elements which have been covered in a previous paper, Motor-Bus Transportation,[2] presented at the 1920 Semi-Annual Meeting, the major issue, of course, is the engine. We employ exclusively the sleeve-valve type. From our viewpoint this type possesses certain basic advantages which make for economy of operation.

First, taking the question of fuel, high gasoline-economy is possible due to

(1) Absence of valve pockets and the spherically shaped combustion-chamber. Incidentally, this permits of high compression being employed. The illustration of the combustion-chamber (Fig. 23) brings out this point very clearly
(2) Positive action of valves at all speeds
(3) Extraordinarily low friction-horsepower
(4) Ideal location of the spark-plug

Next, there is the question of service. In this respect we believe the sleeve-valve engine has the following advantages:

(1) The performance remains reasonably constant throughout the useful life. It is not necessary to make adjustments constantly to permit of satisfactory and uniform behavior
(2) Throughout the useful life the performance tends to improve
(3) Practically no adjustments can be made since there is nothing to adjust. This alone represents a considerable saving in the garage force
(4) Throughout useful life there is little, if any, increase of noise due to wear
(5) Cost of repairs is small since there are very few operations requiring skill
(6) Cylinders never require reboring. This obviates the necessity of carrying in stock second-standard pistons and rings

From Fig. 24 it will be seen that the engine has an exceedingly clean appearance.

The performance of a correctly designed engine is largely a function of its carbureter; therefore a wide variety of results is always obtainable with varied settings. From the graph showing fuel and power output reproduced in Fig. 25 it will be noticed that the characteristics of the sleeve-valve engine are rather remarkable. The setting in question is considered as being particularly suitable for type-J equipment. The points brought out in Table 6 are of special interest.

TABLE 6—HORSEPOWER AND TORQUE DATA FOR TYPE-J BUS

| | |
|---|---|
| Power Developed at 1,000 R.P.M., hp. | 36.20 |
| Power Developed per Cubic Inch of Displacement, hp. | 0.12 |
| Weight of Vehicle per Horsepower, lb. | 301.00 |
| Weight of Vehicle per Cubic Inch of Displacement, lb. | 36.20 |
| Maximum Torque, lb-ft. | 194.00 |
| Speed for Maximum Torque, r.p.m. | 800.00 |
| Decrease in Torque at 400 R.P.M., per cent | 5.10 |
| Decrease in Torque at 1,400 R.P.M., per cent | 11.90 |
| Speed for Maximum Torque with a 5.4 to 1 Rear-Axle Ratio, m.p.h. | 16.10 |
| Minimum Fuel-Consumption, lb. per b. hp-hr. | 0.55 |

Expressing the results obtained in another manner, it is interesting to reflect on the fact that during 1921 our entire fleet of buses averaged 50.7 ton-miles per gal. In connection with the rather remarkable performance which this type of engine delivers in our service, particularly from the standpoint of fuel economy, mention should be made of the carbureter, which is of the Zenith type. From Fig. 26 it will be seen that there is no exterior adjustment. The throttle spindle is 7/16 in. in diameter, hardened and ground. There is a total of 4 in. spindle bearing-area. There is a gland with a suitable packing at the front end and a blank nut at the other. It is interesting to compare the arrangement with conventional designs that in many instances have throttle spindles resembling closely wire nails. With the bus there is an abnormal amount of throttle movement, and unless this factor is taken into consideration from the standpoint of design, rapid spindle and bearing wear will take place. It will also be seen that the design is rugged throughout. All screws, nuts, plugs or unions are of ample size. The butterfly is exceedingly well fitted and provision is made for a simple throttle-stop adjustment. These points are clearly brought out by Fig. 26.

[2] See Transactions, vol. 15, part 2, p. 143.

### Minimum Weight

It seems scarcely necessary here to argue as to the desirability of light weight. These remarks particularly apply to the matter of unsprung weight. Assuming good design, obviously minimum weight means minimum fuel-consumption, maximum acceleration and speed and minimum costs for repairs and renewals. These are the controlling elements. Henry Ford started out with this idea firmly imbedded in his mind and, as far as we know, he has had no cause to change his views.

Clearly, the lighter the vehicle, the easier the solution of our problems. Heavy vehicle-weight means unnecessarily large tires, stronger axles and frame, larger brakes, slower gear-ratios and, last but not least, more engine power. The entire theory of design should be based on the highest safe vehicle-speed for the smallest throttle-opening, and consequently the minimum number of engine revolutions. Of course, this is out of the question if we start off with an unnecessarily heavy unit.

From our experience in operating 21 different types of buses in the past 14 years, we believe that the weights and percentages of axle-load distribution given in Table 7 make for safe and efficient practice.

### Maximum Safe Speed

The greatest single factor from the standpoint of economical operation is speed. This point is perhaps not sufficiently recognized. The following facts in connection with our operation may make the matter somewhat clearer. During 1921 we spent in platform payment, drivers' and conductors' wages, in round figures, \$1,625,-000. So, for each 1-per cent economy in speed there is a yearly potential saving of more than \$16,000. Looking at the situation another way, the ratio of expenditure

between our platform payment and all money expended in connection with repairs and renewals to chassis and bodies, is approximately 5 to 1.

From this it is clear that, while there are always opportunities to effect a saving in connection with maintenance methods generally, the real solution is to employ the fastest possible safe speed and to drive the vehicles up to the limit of their endurance. This, of course, necessitates all that is best from the standpoint of design. Naturally, to maintain a high average rate of speed, rapid acceleration is essential. But in connection with this matter it is well to bear in mind that there is nothing gained and much lost if the engine power is in excess of actual requirements, for it is bound to be abused. A very real problem is to ascertain with each operation the exact amount of power required, then to adopt a standard carbureter-setting with a view to its proper control. Obviously, the questions of acceleration, deceleration and maximum safe speed are closely allied. Reference has been made to deceleration under the heading Effective Brakes.

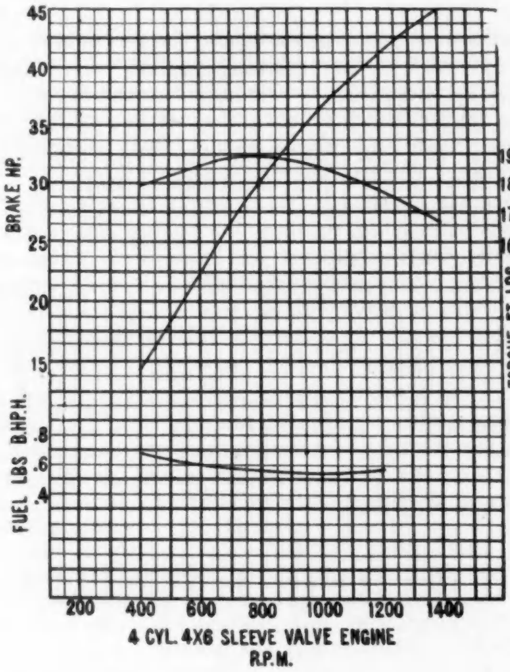



Fig. 25—Curves Showing Engine Performance

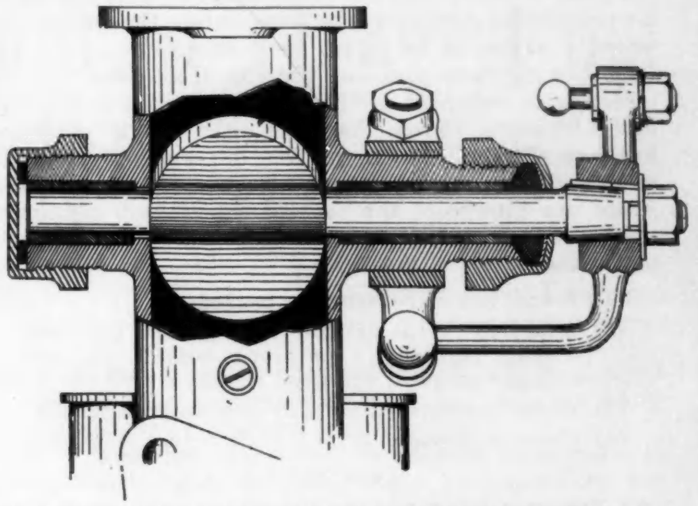


Fig. 26—Sectional View Through the Carbureter Throttle of the Types-J and L Engine

### Maximum Tire-Mileage

In the earlier days of bus operation, the tire question was one of our chief anxieties. Today the situation is very different, for wonderful improvements have been made in tire manufacturing methods. Of course, there is no sense in decreasing tire expenditures at the cost of the equipment generally. Resilient tires are essential and

TABLE 7—DISTRIBUTION OF WEIGHT IN 51-PASSENGER DOUBLE-DECK AND 25-PASSENGER SINGLE-DECK BUSES

| Description | Double-Deck Bus | | | | | Single-Deck Bus | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Total, Lb. | Front | | Rear | | Total, Lb. | Front | | Rear | |
| | | Lb. | Per Cent | Lb. | Per Cent | | Lb. | Per Cent | Lb. | Per Cent |
| Chassis, Gasoline, Oil, Water and Foot-Boards | 6,000 | 3,000 | 50 | 3,000 | 50 | 5,000 | 2,600 | 52 | 2,400 | 48 |
| Body, Sign, Battery and Heaters | 4,000 | 880 | 22 | 3,120 | 78 | 2,000 | 500 | 25 | 1,500 | 75 |
| Chassis and Body | 10,000 | 3,880 | 39 | 6,120 | 61 | 7,000 | 3,100 | 44 | 3,900 | 56 |
| Passengers and Crew at 150 Lb. Each | 7,950 | 2,385 | 30 | 5,565 | 70 | 3,900 | 430 | 11 | 3,470 | 89 |
| Grand Total of Fully Loaded Bus | 17,950 | 6,265 | 35 | 11,685 | 65 | 10,900 | 3,530 | 33 | 7,370 | 67 |
| Load on Each Tire | .... | 3,133[a] | .... | 5,842[b] | .... | .... | 1,765[a] | .... | 3,685[c] | .... |
| Unsprung Weight, Springs to Tires | .... | 700 | .... | 2,000 | .... | .... | 650 | .... | 1,550 | .... |
| Chassis Load on Each Spring | .... | 1,150 | .... | 500 | .... | .... | 975 | .... | 425 | .... |
| Body Load on Each Spring | .... | 440 | .... | 1,560 | .... | .... | 250 | .... | 750 | .... |
| Passenger and Crew Load on Each Spring | .... | 1,193 | .... | 2,782 | .... | .... | 215 | .... | 1,735 | .... |
| Total Load on Each Spring | .... | 2,783 | .... | 4,842 | .... | .... | 1,440 | .... | 2,910 | .... |

[a] Size 34 x 4 in. single.
[b] Size 34 x 5 in. dual. Figure is for each set.
[c] Size 34 x 6 in. single.

too great a wear must not be permitted. It is our regular practice to remove a tire immediately the rubber has worn to within ⅞ in. of the hard base.

In looking back over our records, it is extremely interesting to note that in 1911 our cost per mile for tires was 4.93 cents. From that date on, a steady reduction has been effected. The figure for 1921 was 0.87 cents per mile, and this, of course, includes the use of six tires. From our viewpoint the factors which have permitted this condition to be reached are, in the order of their importance

(1) Better tire manufacturing methods
(2) Improved vehicle design. This includes decreased weight, particularly unsprung weight, the substitution of metal for wood wheels, etc.
(3) Closer supervision from an operating standpoint
(4) Closer supervision from a maintenance standpoint

### Conclusion

As the result of long experience in connection with the design, construction and operation of buses, we are convinced more than ever that trucks or automobiles, modified or unmodified, are absolutely incapable of giving satisfactory and economical service if operated as buses. The tendency today is to employ trucks or automobile chassis as buses, or to attempt to modify their construction, then to re-christen them. This is a dangerous policy from the standpoint of both the builder and the user, and eventually it must surely result in dissatisfaction and disillusionment.

There is another and very important matter: We must not lose sight of the fact that the bus has not made good in some of the localities where it has been tried out. We are constantly confronted with failures such as those at Des Moines, Toledo, Kansas City, and other cities. Such failures, when analyzed, invariably point to the fact that the combination of extemporized equipment, indiscriminate operation, overloading and lack of experience is responsible. But these failures can be avoided, and the automotive industry in its own interest should do all that is possible to guard against such occurrences.

It seems scarcely necessary here to comment upon the splendid achievements of the Society in connection with standardization work in general. Certainly, this has been a controlling influence in the development of the automotive industry. We believe much would be gained if it should now concentrate upon the motorbus. What we have in mind is the standardization of certain of the main dimensions; for example, front and rear-axle tracks, spring center-to-center distances, frame width, dimension between dash and wheel pocket, seat dimensions, aisle widths, etc., for the various classes of service.

The main object of this paper is to bring to the attention of interested parties in a clearcut, vigorous and interesting manner, the fact that to produce motorbus chassis that can be operated efficiently and economically, a very close study must be made of the entire situation. It is also desired to destroy as far as possible the illusion that a bus chassis is merely a modified truck. If in these things, even a moderate degree of success is achieved, we shall feel amply repaid for our efforts.

The matter of body design has not been touched upon since this is a subject that, because of its magnitude, must receive separate treatment.

# PRESIDENTIAL ADDRESS OF B. B. BACHMAN

(*Concluded from page* 12)

creasing traffic, particularly over main routes, will bring a reaction unless we are peculiarly alert to study and suppress in design all objectionable characteristics of our vehicles to the greatest possible degree. I appreciate that the control of all these features is not in the hands of the engineer or builder, but he should be thoroughly posted as to what they are and be prepared to cooperate intelligently with regulatory bodies to assure that rational measures for the protection of the public, which do not impose unreasonable restriction on road transportation, are enforced.

Originally road construction was in the hands of individuals or corporations that operated them for profit in the collection of tolls. While we have rejected, as a Nation, the idea of public ownership of the railroads, so also have we rejected the idea of private ownership of the highways. I believe both these ideas are proper. In the railroad we require concentration of authority and responsibility in operation over any one given line. This can be obtained most efficiently by private ownership and operation under reasonable government regulation. The highway, on the other hand, is primarily for the use of the individual according to his needs and desires, with as little restriction as possible consistent with public safety, which can be obtained best by public ownership and complete government control through one of its departments.

Much of the discussion on the question as to who should bear the burden of the cost of construction and maintenance of our highway systems, or whether the motor-vehicle operator is receiving a public subsidy that is not shared by the railroad, etc., appears to be beside the point. The cost of transportation of passengers and freight, by railroad, water or highway, is borne by the whole community and shared by every citizen in proportion to his requirements for transportation. I believe this to be so, whether the cost of transportation is included in the cost of the commodity or it appears partially in the form of taxes. The big fundamental problem is to determine the economic field for each medium of transportation and the relation each should bear to the other for maximum efficiency, and the most satisfactory means of proportioning the expense to the individual.

I have endeavored to the best of my ability to give you a brief and yet comprehensive view of the problems that are confronting us and should receive our active individual and collective attention. I hope the result may be to stimulate interest in the affairs of the Society and enlargement of the horizon in our view of the future activity of, and the service that can be rendered by, each of us individually and as an organization.

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# The Hot-Spot Method of Heavy-Fuel Preparation

*By* F. C. Mock[1] and M. E. Chandler[2]

Semi-Annual Meeting Paper *Illustrated with* Drawings

THE development of intake-manifolds in the past has been confined mainly to modifications of constructional details. Believing that the increased use of automotive equipment will lead to a demand for fuel that will result in the higher cost and lower quality of the fuel, and being convinced that the sole requirement of satisfactory operation with kerosene and mixtures of the heavier oils with alcohol and benzol is the proper preparation of the fuel in the manifold, the authors have investigated the various methods of heat application in the endeavor to produce the minimum temperature necessary for a dry mixture.

Finding that this minimum temperature varied with the method of application of the heat, an analysis was made of the available methods on a functional rather than a structural basis. Three of these are discussed: (*a*) When the heat from the walls of the manifold is applied through the medium of the air; (*b*) when it is applied to the fuel alone, or partly to the fuel and partly to the air; and (*c*) when a spray of atomized fuel and air is directed against a heated surface. A device was constructed by which the three main variables, the exhaust temperature, the exhaust flow and the area of the heating surface, might be regulated and the three remaining variables, the quantity of air, the quantity of fuel supplied and the quantity of fuel vaporized, might be controlled.

Taking into account the wide range of temperatures that the air charge and fuel supply undergo before entering the intake-manifold system, a quantitative computation of heat transfer was made and the conclusions were drawn that only by a combination of centrifugal force, surface tension and the force of gravity could the unvaporized drops be separated from the fuel charge and that the conditions of combustion are governed by the rate of fuel feed from the manifold to the cylinder and not from the carbureter to the manifold.

RECENT years have witnessed increasing attention to the design of intake-manifolds and to the varied methods of handling automotive fuels in preparing them for introduction into the combustion-chamber. The resulting development, however, has been limited in direction, being confined usually to slight modifications of the construction that has been followed ever since automotive engines began to have more than one cylinder. The improvement in economic conditions that all authorities agree is approaching will certainly result in considerably increased use of automotive equipment, and it is not impossible that the demand for motor fuel may bring about a condition of higher cost and lower quality. As it might require three or four years to develop a change of design to meet such a change of fuel, it would seem that now is a fitting time to make a survey of the problems involved in the preparation of our present fuels and of heavier ones and to make a fresh analysis of the situation, entirely apart from and unhampered by the conditions of previous practice.

It is true that many 1922-model cars have operated satisfactorily with the motor fuels at present in use, both in summer and in winter, but many have not. We are convinced that it is possible to operate on mixtures of gasoline, kerosene and some heavier oils, combined with alcohol, benzol or other anti-knock component, as well or better than a number of cars today operate on gasoline, by the use of improved methods of fuel preparation in the intake-manifold.

## Specific Requirements of Fuel Preparation

The requirements of proper fuel preparation are

(1) A thoroughly and continuously homogeneous mixture of fuel and air with no drops or liquid-film wall-flow to the valve ports
(2) The charge temperature should be the minimum possible while complying with requirement (1)
(3) The provision for a prompt change in the rate of action under changes of load and speed

A cylinder charge of fuel is only a medium-sized drop. Any one who has observed through glass manifold sections, the storm of drops that is usually present, can easily appreciate the importance of this point. All the oil dilution in the crankcase is due, of course, to the introduction into the combustion-chamber of fuel that is not burned later. We believe that a large part of the rapid carbon formation, characteristic of engines having poor distribution, is due to the cracking, without burning, of the drops of excess fuel that occasionally enter the cylinders. The first requirement would include the prevention of liquid gasoline from reaching the cylinders after the use of a primer or choke means of starting. As starting is really an increase of the load from zero, devices for starting and quickly warming-up come under the last requirement.

## Methods of Heat Application

If we consider as our objective a minimum temperature of the dry mixture, that is, a mixture of transparent fuel-vapor and air, it is immaterial, in theory, whether the heat is applied first to the fuel or to the air. If, however, we accept what our experiments have apparently demonstrated, that is, that a fog mixture of condensed vapor and air is satisfactory, provided the cylinder temperatures are such as to change this fog to a vapor before the end of the compression stroke, we shall find, both in theory and in practice, that the minimum temperature that can be used will vary with different methods of heat application. The theoretical considerations involved are, we hope, clearly shown by an analysis of the known and available methods of heat application. These have been classified as follows:

Case No. 1. Heat imparted to the mixture through the medium of the air, by the communication of

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[2] M.S.A.E.—Engineer of carbureter design and development, Stromberg Motor Devices Co., Chicago.

heat from the manifold walls to the air and to such part of the fuel as has been deposited on the manifold walls. This is considered to involve only the production of a dry-vapor mixture. An interesting variation of this is shown as Case No. 1b, where part of the preheating of the air is accomplished by subtracting heat from the air and the vapor mixture already formed, thus giving a fog mixture

Case No. 2. Application of heat first to the fuel alone, with resulting condensation of the vapor when it joins the main air-column; this results in a fog mixture

Case No. 2a. Heating the fuel and a part of the air to generate a rich dry-vapor mixture, which is then condensed as it enters the stream of the remaining air-supply. This gives a fog mixture

Case No. 3. Directing a spray of atomized fuel and air against a heated surface. One result obtained with this construction is the breaking-up of the spray drops into even smaller drops in the so-called "spheroidal" condition; the mixture thus formed can scarcely be properly designated as a fog mixture

This classification has been made on a functional rather than a structural basis. Most of the hot-spot constructions in actual use employ two, and sometimes three, of these heating methods, but for analysis the distinction we have made seemed necessary. Consideration of the direct application of heat to the fuel has been purposely limited to designs in which the fuel has been previously metered in a liquid state, as doing so after heating has not thus far been demonstrated as practicable.

The computation of the mixture-temperatures is based upon the methods used and determinations made by Professor R. E. Wilson and described in THE JOURNAL.[3] The gasoline values used are those of the high end-point gasoline referred to in Professor Wilson's discussion in THE JOURNAL[4] for October 1921. This gasoline, by the way, is apparently quite similar to the "D" gasoline of the fuel research consumption test recently concluded.

[3] See THE JOURNAL, November 1921, p. 313, and January 1922, p. 65.
[4] See THE JOURNAL, October 1921, p. 265.

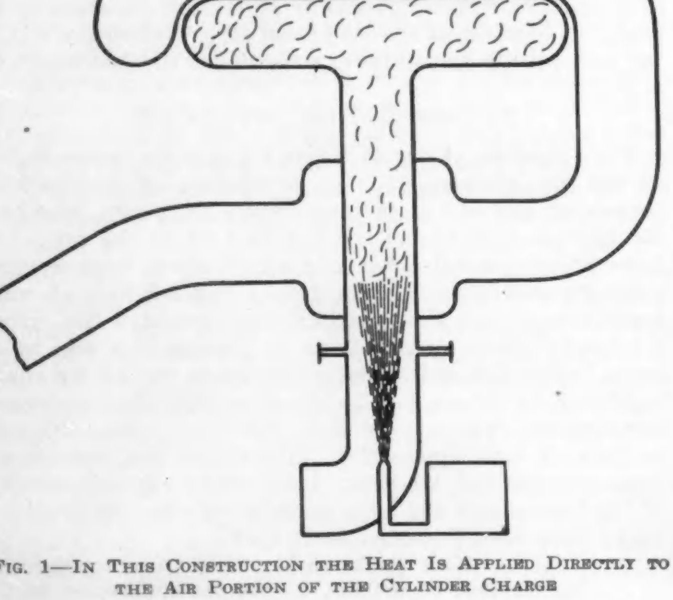


FIG. 1—IN THIS CONSTRUCTION THE HEAT IS APPLIED DIRECTLY TO THE AIR PORTION OF THE CYLINDER CHARGE

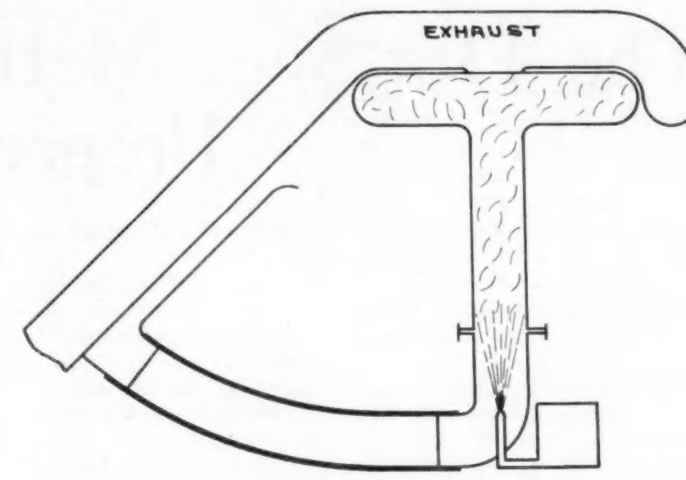


FIG. 2—MANIFOLD CONSTRUCTION TO SUPPLY HEAT TO THE AIR CHARGE BEFORE IT ENTERS THE CARBURETER

## HEAT APPLIED DIRECTLY TO THE AIR

In Case No. 1 the air charge receives a heat supply such that after the latent heat of evaporation has been supplied to the fuel, the resulting mixture will have the minimum temperature of a dry vapor. A typical construction is shown in Fig. 1. Its practical equivalent, the application of heat to the air charge before it enters the carbureter, is shown in Fig. 2. With the complete evaporation of a 15-to-1 mixture of the high end-point gasoline measured by Professor Wilson, and ignoring the heating of a small amount of fuel on the walls, this would involve air entering the carbureter at 181 deg. fahr. with a resulting mixture-temperature of 135 deg. fahr. For the kerosene measured by Professor Wilson, the air would have to enter the carbureter at 283 deg. fahr. with a final mixture-temperature of 230 deg. fahr.

In practice, however, dry mixtures are not realized at such low temperatures, for the reason that only part of the hot air comes into contact with the fuel. Within a short distance from the carbureter jet the tiny droplets of fuel spray take up a velocity and direction identical with that of the air which bears them and thenceforth, until they strike a wall, they generally are surrounded by a miniature atmosphere of vapor at the dew-point. Fuel that travels along the walls comes into actual contact with only a thin film of air. We have endeavored by various means to create a turbulence that would accelerate and decelerate the spray droplets in the air medium that carries them, but every effort of this kind has resulted in increased deposition of fuel on the manifold wall and has made conditions worse than before. The temperatures actually existing in practice are more nearly those that would result if the fuel came into heat-conducting contact with but one-half to one-third the air charge during the travel through the intake-manifold. On such a basis the average temperature of the mixture is considerably higher, for instance, with a 15-to-1 dry mixture, and, if the fuel receives heat from one-half the air, the final average temperature will approximate 175 deg. fahr. with gasoline and 276 deg. fahr. with kerosene, as is brought out in Table 1.

On account of the high heat-capacity of dry-mixture charges formed in this way, there being no cooling from any further evaporation of the fuel during the compression-stroke, the tendency toward detonation should be, and apparently is, greater with this method of fuel preparation than with most others. Due to the relatively slow heat-transfer, more than the customary difficulty is expe-

rienced during changes of engine speed and load. The proper functioning of a device of this kind is contingent upon the maintenance of adequate temperatures; but in actual practice such temperature regulation is disturbed by a number of factors, depending upon seasonal and climatic conditions, as will be explained later. Since the mixture-temperature depends upon that of the air entering the carbureter, which in most cars depends in turn upon the temperature of the cooling water and of the whole mass of metal under the hood, there is a long duration of "warming-up" which can be taken care of only by elaborate thermostatic devices. A factor of safety, to provide for the occasional use of fuels heavier than the average, can be obtained only by raising still farther the temperature of the fuel charge of normal operation.

TABLE 1 — FINAL AVERAGE MIXTURE-TEMPERATURE WITH ENTERING AIR AND FUEL AT 75 DEG. FAHR.

| Fuel | High End-Point Gasoline | | Kerosene | |
|---|---|---|---|---|
| Air-Fuel Ratio | 12 to 1 | 15 to 1 | 12 to 1 | 15 to 1 |
| When Heat Is Transferred from All the Air to the Fuel (Fig. 2) | 145 | 135 | 240 | 230 |
| When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 2) | 195 | 175 | 299 | 276 |
| When Heat Is Transferred from One-Half the Air to the Fuel (Fig. 3) Followed by Cooling of the Charge by the Intake Air | 140 | 128 | 196 | 176 |
| When Heat Is Applied Directly to the Fuel (Fig. 4) | 143 | 132 | 174 | 159 |
| When the Fuel Only Is Heated to Temperature of Required Vapor Density (Fig. 6) | 98 | 92 | 124 | 116 |

More important is the fact that there is nothing to prevent raw gasoline entering the cylinders during the starting and warming-up period and probably also during normal running.

In Case No. 1b, Fig. 3, the fuel vapor is formed as in Case No. 1, but a smaller exhaust air-heater is used. The air entering the intake system, before it reaches the exhaust heater, is used to cool and condense to a fog the dry mixture coming from the carbureter. The temperatures of the air entering the carbureter and of the mixture leaving the carbureter are the same as in Case No. 1, but the final mixture-temperature in the intake-manifold, if a complete heat-transfer could be established, would be considerably lower than in Case No. 1; for instance, 128 deg. fahr. with gasoline as against 175, and 176 deg. fahr. with kerosene as against 276. But we do not believe that in practice the addition of this condensing device would be of value. If made elaborately enough to accomplish the desired heat-transfer, it would probably increase the amount of fuel on the walls and require a still higher temperature of the air entering the carbureter. It would also increase the difficulties of acceleration and the "loading" in the intake-manifold while the engine is cold.

### Heat Applied Directly to the Fuel

In this method, which is shown in Fig. 4, the fuel, after being metered is discharged into a heating chamber that the air charge does not enter; the vapor formed here is then mixed with the unheated air-charge to form a true fog-mixture. At first thought this system seems to be promising, but actually it has serious inherent disadvantages, for the reason that the delivery of vapor depends upon the temperature being kept above a certain minimum.

A homely illustration of the difficulty of evaporation with this type of heater is afforded by the example of a covered kettle or pot of water maintained at a temperature slightly below the boiling point, say 208 deg. fahr. As any housewife knows, a kettle can be heated in this manner for a long time without losing much water, the reason being that, although the evaporation from the surface of the water is rapid for a while, until the space above the water and beneath the lid becomes filled with vapor, there is no difference in pressure between the vapor and the outside air and no marked escape of water vapor from the spout. It is only when the temperature is

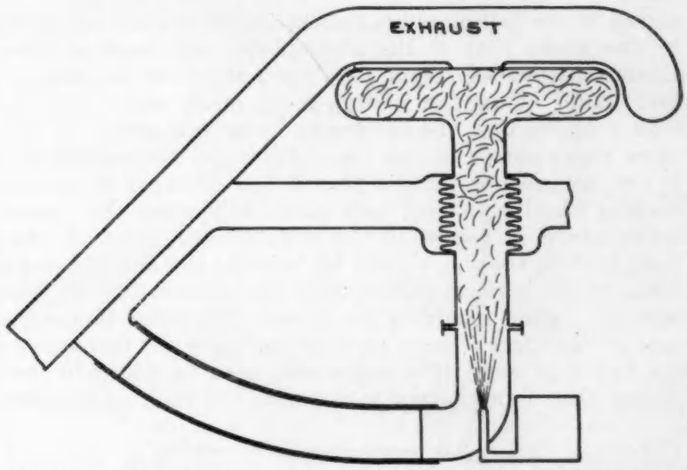


Fig. 3—In This Construction Part of the Initial Heat Application Is Obtained by Cooling the Vapor Mixture to a Fog

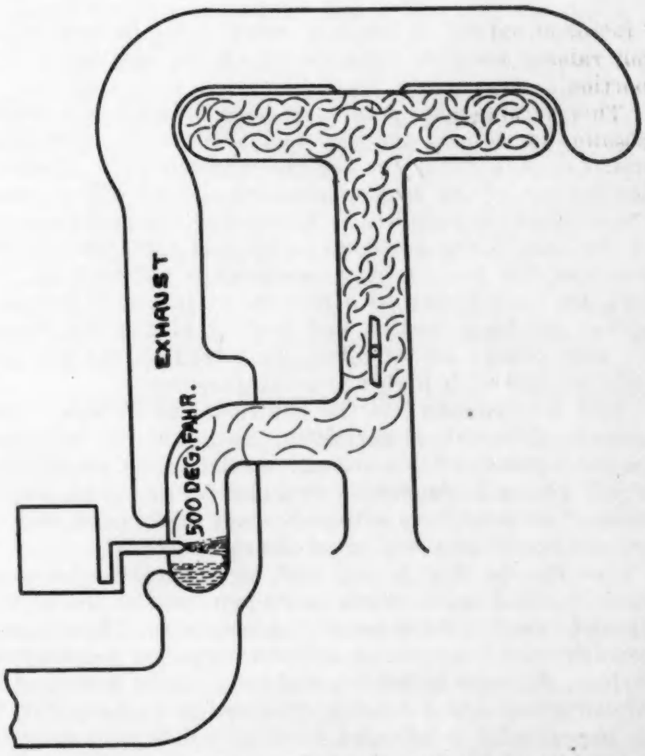


Fig. 4—In This Construction the Heat Is Applied Directly to the Fuel Portion of the Mixture Charge

raised to the boiling-point that the vapor pressure is able to rise above that of the atmosphere and create a continuous outflow of steam. An open chamber will evaporate liquid below the boiling-point much more quickly than a closed one, the difference being due solely to the more rapid escape of the vapor from the open chamber. In the design illustrated a normal flow of vapor from the heating chamber should take place only when the vapor temperature is raised to the final boiling-point of the fuel; that is, the vapor must be between 400 and 500 deg. fahr., which is much higher than the temperature needed with any other construction shown. The final temperature of the mixture may, however, be quite low because of the fact that very little more heat need be added to the system than is necessary to vaporize the fuel. Also, the "factor of safety" in heating capacity may be large without raising the final temperature of the mixture in proportion.

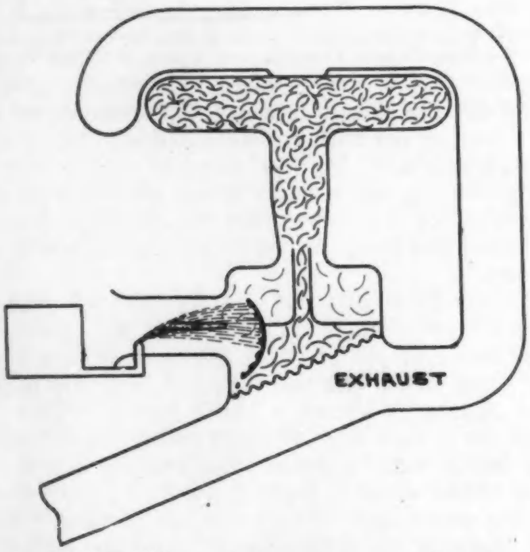


FIG. 5—IN THIS MANIFOLD ARRANGEMENT THE HEAT IS APPLIED DIRECTLY TO THE MAIN PORTION OF THE FUEL AND A SMALL PORTION OF THE AIR

This arrangement might be hard to start and would possibly be slow on acceleration. With heavy fuels there would be a tendency for the heavy elements to collect in the bottom of the heating chamber during idling, when the exhaust temperature is lower than the boiling-point of the fuel. Upon a sudden increase of the exhaust temperature this pool of heavy elements is apt to coke. In fact, we have known of a number of instances where a pocket for the collection and heating of the fuel would fill with greasy tar or coke. This trouble was particularly marked with high end-point gasolines.

This construction has the additional advantage, when properly designed, of permitting no liquid fuel to reach the valve ports; on this account, as also with Case No. 2a, it will give a homogeneous fuel charge, or "good distribution," as we call it, with any shape of intake-manifold and any convenient location of the carbureter.

Case No. 2a, Fig. 5, is a sort of compromise between Cases Nos. 1, 2 and 3, which seems to possess all the advantages of Case No. 3 and fewer disadvantages. The mixture spray from the carbureter is thrown against a deflecting surface, that may be heated, and the fuel not vaporized is thrown down into a heating chamber as in Case No. 2. An opportunity is afforded for the fuel to evaporate in and mingle with the air, before the separation of the liquid and the vaporized portion. This reduces the fuel lag on acceleration and also reduces the amount of fuel that must be taken into the heating chamber. An air circulation is maintained through the heating chamber, which helps to carry the vapor away as fast as it is formed; the action in the heating chamber then can be *evaporation* rather than *boiling* as in Case No. 2. This distinction is important because *boiling* implies the maintenance of temperature above a certain point, at all engine speeds and at a constant pressure, while evaporation can take place at any temperature and, fortunately, under a change of engine speed, the decrease of the exhaust temperature is accompanied by a reduction of the fuel feed and the rate of evaporation required.

The air taken through the heating chamber is, of course, highly heated, so that, as compared with Case No. 2, we have a small part of the fuel and of the air heated, to be cooled by the remainder of the air charge and a certain part of the fuel charge. The temperature balance would, of course, depend on the percentages of the fuel initially vaporized and of the air passed through the heating chamber.

This arrangement possesses the advantage of Case No. 2, in allowing a large reserve capacity for warming-up without excessive heating of the mixture under normal operation, and also of preventing liquid fuel from going into the engine cylinders. A device of this sort, though of design entirely different from Fig. 5, has been used in the actual driving of a passenger car with a six-cylinder engine and gave as good a demonstration on kerosene, with a benzol component to avoid detonation, also alcohol at a mixture-temperature of 120 to 140 deg. fahr., as with gasoline. It was also found possible to use heavier fuel combinations which resulted in perhaps better operation than that shown by the average car in the hands of its owner: One of these mixtures was one-third benzol, one-half kerosene and one-sixth Mobile B lubricating oil; another one-fifth alcohol and four-fifths 38 to 40-deg. Baumé distillate, a light oil that cannot be ignited by itself with a match in the atmosphere at ordinary temperatures and which will burn slowly from a wick with a very smoky flame. With these latter mixtures the mileage per pound was not as good as with gasoline or kerosene, and there was a perceptible carbon-deposit; also a slight slowness, but not hesitation, on acceleration. The operation of the car in general was so good that it would easily satisfy the average car-owner, were it not for the necessity of starting on a different and lighter fuel. Starting on gasoline in very cold weather was not more difficult than with the ordinary carbureter and intake-manifold arrangement. In fact, no difficulty was ever experienced in starting; the starter was always strong enough to turn the engine over, and closing the choke would always effect a start. On gasoline the warming-up was very good. In weather 10 deg. fahr. above zero, it was necessary only to use the dash mixture-control device for about ½ min. or less after starting, after which it was possible to set all the controls in the normal driving position and drive away. This usually synchronized with the development of a mixture-temperature of about 90 deg. fahr. With gasoline the fuel-consumption was but slightly lower on a gallon test than with a good carbureter on a conventional type of hot-spot intake-manifold, but the engine would run smoothly on very lean mixtures and the weekly mileage, particularly in winter, was better. The smoothness and the absence of carbon, crankcase-oil dilution and ignition trouble were marked. We found also improved operation at low speed on hills. The engine would pull smoothly and without apparent

effort and maintain this smooth low-speed pulling indefinitely.

## Air and Fuel Charge Projected Against the Hot Surface

As illustrated in Fig. 6, this includes a condition aimed at, and more or less realized, in many hot-spots in use today. It is the general belief, perhaps, that the fuel spray strikes the heated surface, vaporizes, and then condenses in the airstream. More recent observations lead us to believe that very little of the fuel vaporizes on the heated surface. It seems rather that the sudden application of heat to one side of the drops of spray, as they strike the heated surface, relieves the surface tension that holds them in globular form and causes them to burst; meanwhile, if an air-draft is present, the "spheroidal condition" keeps them from adhering to the heated surface. This belief was first suggested by the observation that large drops come off such a hot-spot in a coarse spray, while small drops come off in a finer spray.

There is one interesting hypothesis of action under these conditions, the realization of which would give a fog mixture at very low temperatures with a very simple structure. If the heating surface were of exactly the size and location to be wholly covered by the liquid of the fuel spray; if its heating capacity were such that it could vaporize all the fuel that strikes it; and if the scouring action of the air-draft across the heating surface were sufficient to carry away the vapor as fast as it was formed, it would be possible to produce the vapor at the relatively low temperature corresponding to a density of one-fifteenth to one-twelfth that of air; also, there should be little, if any, heat transmitted directly to the air from the heating surface. Under such conditions, which we believe can be realized only in theory, the mixture temperatures would be the minimum among all the systems suggested for producing a fog mixture by external application of heat energy.

Fig. 7 is an effort to show the nature of such action, assuming complete evaporation at the surface. There is, first, near the surface, a film of liquid, or a layer of liquid

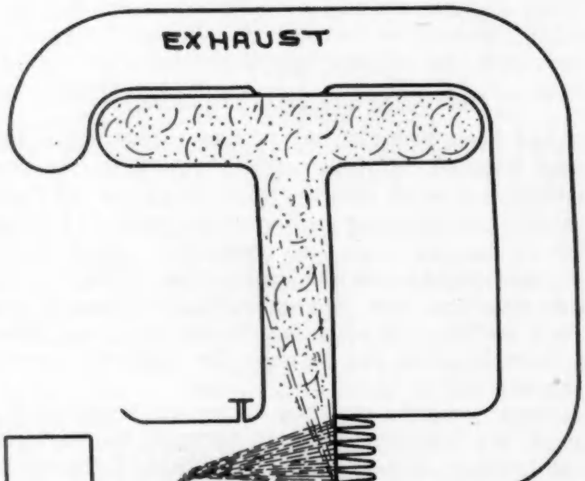


Fig. 6—In This Manifold the Air and Fuel Charge Is Projected Against a Heated Surface

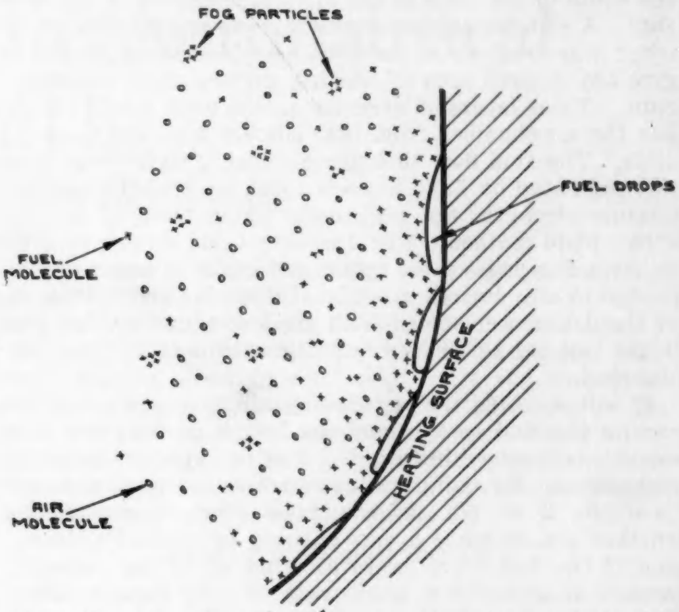


Fig. 7—Diagram Showing the Nature of the Action of the Construction Illustrated in Fig. 6

drops. Just off the liquid film the greatest vapor-density occurs, but as the distance increases and the air begins to lower the temperature, the molecules will begin to gather in droplets, in the action that we term condensation. It is obvious that it would be impossible to bring *all the air* into such contact with the liquid film that the vapor would be swept away, and uniformly diffused, within a few molecule paths of the liquid film; and it is only under such a condition that the temperature balances of Fig. 6 could be obtained. But it also is clear that the more completely we can direct and diffuse the air charge on the heating surface in the conventional hot-spot design, the lower the temperature and density can be next the liquid film, the lower can be the temperature of the liquid film and the wall itself and the lower the final temperature of the charge.

Regardless of the correctness of the theory of operation of this type of hot-spot, there are several advantages and disadvantages in practice that should be pointed out. As already outlined, reserve capacity can be obtained only by making the surface larger. Also, there is no inherent characteristic of this arrangement that would prevent liquid fuel from going into the engine. The heat capacity of the wall of any structure that could be used would be sufficient to prevent any lag in acceleration, provided the carbureter were made to give a charge of slightly increased richness, with a fuel of graduated volatility.

## Experimental Determination of Heating Action

The foregoing analysis indicated the great importance of several considerations not previously investigated in the problem of properly preparing the fuel. We undertook, therefore, to build an experimental device that would allow us to regulate the three main variables governing the heat input; (*a*) the exhaust temperature, (*b*) the exhaust flow and (*c*) the area of heating surface; and to control the remaining variables affecting heat absorption; (*d*) the quantity of air, (*e*) the quantity of fuel, and, so far as possible, (*f*) the vapor density. The device used is shown diagrammatically in Fig. 8. The heating element was an iron plate, 3½ in. wide and about 8 in. long, exposed to the exhaust on the ribbed lower side, and receiving fuel from a series of jets placed across

the width of the plate at the air-intake end of its top surface. A slab of magnesia cement encased in a thin metal cover was used as a movable heat-insulating shield to give any desired area of heating surface up to the maximum. Thick layers of asbestos gasket were used to insulate the air-chamber from heat contact with the heating-plate. The fuel flow took place under gravity head and was regulated by hand at each speed to give the desired mixture-strength, the air supply being metered by the orifice-plate method. The fuel-jets could be set to give either a fine or a coarse spray and could be made to discharge in any desired direction. One side and both ends of the device were fitted with glass windows so that the liquid and fog conditions existing within could be easily observed.

It will be noted that by closing air entrance *a* and directing the fuel spray along the length of the plate, the conditions involved in Case No. 3 of our analysis could be reproduced. By using air entrance *a*, the conditions of Case No. 2 or No. 2a would be given, according to whether air entrance *b* was entirely or partially closed. And if the fuel spray were directed along the passage, instead of at the hot plate, with the air supply taken through entrance *b*, the conditions of Case No. 1 would exist.

The first test-runs with this device indicated that a broader range of investigation than initially planned was advisable. Unfortunately it was impossible to complete the program in time to include its results in the preprint of this paper, but it is hoped that at the Summer Meeting a resume of the observations and measurements of several different fuels can be given, the tests to include measurements of the mixture-temperatures, the heating surface necessary, the variation of temperature with increased heating surface and "volumetric efficiency" under the above variations.

### Natural Variation of Air Temperature Under the Hood

In the quantitative computation of heat transfer, we first must take into account the very wide variations of temperature that the air charge and fuel supply undergo before they enter the intake-manifold system, on account of the large range of variation of hood temperature. Fig. 9 is given as a rough indication of the various changes in temperature that a molecule of air undergoes in getting from the external atmosphere to the cylinder port, without purposeful application of heat to the intake charge, other than the commonly used hot-air stove around the exhaust-pipe. Starting from atmospheric temperature, the temperature of the air is raised between 30 and 60 deg. in passing through the radiator. It has been our observation that there is a greater difference between the motometer temperature and that of the external air in summer than in winter. Perhaps some of the radiator engineers can tell why this is so. In the summer there is sometimes an additional rise of temperature under the hood due to the radiation of heat from the engine. The rise of temperature from the hot-air stove is presumably about the same in the summer and in the winter but on many cars an appreciable portion of the heat added by the stove in the winter is lost before it gets to the carbureter, because of the cooling effect of the fan-blast of relatively cold air on a long length of flexible tubing. The temperature-drop in the carbureter and the manifold due to vaporization is indeterminate, dependent upon the fuel, the temperature and the vacuum. In many cars the intake-manifold is so close to the exhaust that under full load the temperature is raised considerably by the cross radiation. We have sometimes gained 3 to 4 hp. in a maximum of about 70, by cutting off this radiation with asbestos board.

Fig. 9 will give an idea of the range of natural temperature-variation with which our intake systems have had to deal. Between the temperature of the air entering the intake system just after starting in the winter and that during a long run in the summer, there easily may be a difference of 120 deg. fahr. Very few current applications of heat to the intake charge, by either hot air or hot-spots, affect the temperature one half this amount. Any effort to attain minimum charge-temperatures in actual practice must include means for dealing with the natural temperature-variation under the hood.

### Means of Temperature Control

With heating methods that approximate Fig. 1, the heating surface should perhaps be in two sections, one of which is in action at all times, and the other of which may be thrown open to the exhaust, either by a seasonally regulated valve, or by the dash mixture-control of the carbureter.

Arrangements such as that shown in Fig. 2 can be controlled within certain limits of temperature by using a hot-air stove on the exhaust line that has at least three times the heat capacity of those in common use today, with a valve adapted to cut-off part of this hot air and admit cold air as the engine warms-up. The regulation should preferably be automatic.

In the methods of Case No. 2, Figs. 4 and 5, no particular regulation for variation of atmospheric temperature is necessary. The heating action is almost independent of the outside temperature. With this type of construction, I have always recommended making the heater large enough so that cool air from outside the hood can be taken into the carbureter in the summer time.

Fig. 6, like Fig. 1, would perhaps be taken care of best by a regulable variation in hot-spot area. Difficulty is experienced in practice in confining the heat to the region where it is desired. In warm weather the heat from the warm hood atmosphere tends to conduct across the flange junctions and through the walls of the heating chamber. In future designs we may find thick heat-insulating material, or spacers of refractory tile, used to separate the hot from the cool portions of the heater.

### Homogeneous Mixture Quality

As has been brought out in the foregoing, a homogeneous mixture requires a fine spray from the carbureter issuing directly into the heating region. If the fuel is allowed to condense or gather on the walls, it will reach the hot surface in waves and irregularly timed splashes, under which conditions the carbureter setting must always be somewhat rich, and many details of engine operation will suffer. Acceleration is always more difficult when there is a fuel lag between the carbureter and the heating surface.

The arrangement, common in many heavy-duty engines, of locating the governor between the carbureter and the hot-spot, is very bad. Everything indicates that the carbon deposit will be reduced to the minimum and crankcase-oil dilution eliminated only when this custom is disregarded and the carbureter is placed close to the hot-spot.

Several 1922 engines that have the property of operating very smoothly on extremely lean mixtures, have intake-manifolds that are characterized by a hot-spot at the

(*Concluded on page* 48)

# Progress of British Aeronautical Research[1]

*By* Brig-General R. K. Bagnall-Wild

RESEARCH work for the Air Service comprises specific researches at establishments under the control of the Air Ministry and an important series of studies at the universities by arrangement with the Ministry. Much research work is being conducted at the universities independently of the Government, the results of which are of great importance to the future of flying. It is at least as important for the director of research to be in touch with cognate researches in the universities and other research organizations, at home and abroad, as it is for him to be cognizant of the developments aided by State finance, through the Air Ministry. The chief research establishments under his direct control are the Royal Aircraft Establishment at Farnborough, the National Physical Laboratory and the Air Ministry Laboratory. The practical applications of the results of research are tried out at the air-stations at Martlesham Heath, the Isle of Grain, Biggin Hill and, by courtesy of the Royal Air Force Coastal and Inland Areas, at certain other air-stations.

The directorate of research is, however, an engineering as well as a scientific organization. It acts as the engineering department of the Ministry and probably as much as four-fifths of its work relates to experimentation with specific appliances, efforts to develop such appliances along channels useful to aeronautics, and test performances of approved air-service material. The remainder is research, pure and applied.

Long experience has taught the universities the most suitable organization by which research as distinguished from ordinary technical work can be undertaken. This requires for its success a much greater personal freedom of work than had previously been the rule in Government organizations. In the last Wilbur Wright lecture before the Royal Aeronautical Society Major G. I. Taylor pointed out the great difficulty of organizing scientific research on the man-hour principle, adding that in his opinion research work is so difficult and exacting that a man can do his best work only when he is free to go where his researches lead, free to choose his own time for work, free from other duties which would divert his mind and lastly, free to sit down and produce no visible result for an indefinite period. In certain forms of applied research the problem is little different, and this is one phase of the work with which the directorate of research has to deal. The other, larger, but fortunately simpler, portion of the work consists in the design, development and ultimately the production of apparatus or methods which the work of the research organization shows to be capable of serving some definite purpose. In theory it may be possible to draw a sharp dividing line between these two classes of work, but in practice they tend to shade into each other in a bewildering way, and such boundary as may develop is liable to vary its position in accordance with the subject under investigation. With limited funds it is sometimes necessary for the research worker who may have started a new development, to see it through its infant difficulties, and mother it until it is in suitable form for the production stage.

It may, I think, be stated fairly that the biggest technical problem affecting civil aviation is the developing and perfecting of the airplane engine; the nature of the ground organization, the question of flying over or under clouds, the attractiveness of air travel to possible passengers and numerous other questions turn on the fundamental issue of engine performance.

### Aeronautic Engine Research

The ultimate aim of aeronautic engine research is to produce an engine as free from breakdown as the average engine in a motor vehicle. It must be recollected, however, that the engine in a motor vehicle runs usually at about one-third its full brake-horsepower, whereas until recently it was customary for aircraft engines to run at from 80 to 100 per cent of their full-load capacity; it is only now, with the Napier Lion engine, that on the cross-Channel service it is possible to run with but 60 per cent of the maximum load. This, in my opinion, is the fundamental difficulty. This is, of course, much increased by the need for reducing the weight per horsepower to a minimum.

Many pin their faith to the almost magic properties of specific fuel-mixtures, but the work of Tizard and Pye and of the Ricardo Laboratory has shown that the efficiency and horsepower that are obtained from any mixture of the various volatile hydrocarbon fuels are within 2 or 3 per cent the same as from any others, provided the compression-ratio is not altered. Any advantage to be gained from a specific fuel-mixture must lie in its relative freedom from detonation, and therefore in its suitability for employment at a higher compression-ratio, with the resulting higher fuel-economy.

Experiments are being made to determine whether it is better to replace the carbureter system by some method of direct injection.

The experimental flight carried out by my predecessor, from Egypt to Mesopotamia and back, brought out with much emphasis the need for economizing in water on such flights. If water could be dispensed with entirely in favor of air-cooling a great step forward would be made. From a fighting point of view also this would be of great advantage since it would make the engine much less vulnerable to hostile machine-gun fire. The staff at Farnborough have long been of the opinion that this is a possible development, and we now have the Bristol Company's Jupiter engine of 380 b. hp., a nine-cylinder radial engine of which much is expected. There is also the Siddeley Jaguar, of 350 hp.; while in some other tests a single air-cooled cylinder has given as much as 100 hp., with a brake mean effective pressure of 134 lb. per sq. in. Some enthusiasts consider that an air-cooled engine of 1000 hp. should not be impossible of attainment; and that with the necessary cooling equipment an aeronautic engine of 2400 hp. at 750 r.p.m. may be realized. An advantage of large engines is that the lower number of

[1]From an abstract of a paper read at the 1922 Air Conference held at London. The author is director of research for the Royal Air Service.

revolutions per minute avoids the use of reduction-gearing to the propeller and reduces the weight about ½ lb. per b. hp.

When comparisons are made between the weights of air-cooled and water-cooled engines, it is sometimes forgotten that, although in the former case the weight of radiator and water is saved, it is necessary to include on the other side of the ledger an item for the additional weight of the air-cooled cylinders. In a comparison made recently it was found that although the gain due to the elimination of radiator, water, pipes and pump was about 0.70 lb. per b. hp., against this was an excess in cylinder and piston weight of 0.46 lb. per b. hp., so that the net gain was only 0.24 lb. If the fuel and oil consumption of the water-cooled engine be put at 0.50 lb. per b. hp-hr. and that of the air-cooled engine at 0.58 lb., the two become equal as regards overall weight when a 3-hr. supply of fuel and oil is included.

### Navigation

The most difficult of all problems in connection with "navigation" is the provision of means to enable an aircraft pilot to locate the airdrome for which he is bound in foggy or misty weather and to make a successful landing. Unquestionably it is this difficulty that causes pilots on long routes to prefer to fly under, rather than over, clouds.

When in 1912 use was made of the precessional movement of a gyrostat to measure the velocity of roll on one of His Majesty's ships, it was not contemplated that by far the most successful application of the apparatus would be for air travel; it is known as the gyro-turning indicator. Ample experience has shown that it is more sensitive and much more rapid than any other method of indicating the turns of an airplane. Unlike a constant-azimuth *gyro*, this apparatus does not need to be delicately balanced as regards the position of its center of gravity and is therefore remarkably foolproof. Airplanes flying in a fog frequently get into turns without knowing it. When they do so, the magnetic compass, particularly when of an old pattern, is reasonably sure to indicate a turn in the wrong direction, and to thoroughly mislead the pilot.

### Machines

The question of stability is being continually studied. Stability may be either inherent or automatic. Automatic stability is attained through the operation of some more or less complicated auxiliary mechanism, while inherent stability is due directly to the nature of the design of the aerodynamic surfaces, the disposition of weights and similar factors. If inherent stability could be attained there would be small need for automatic stability. The problem of longitudinal stability is to a large extent solved, but that of lateral stability will require much work.

Attention has been drawn to the interesting arrangement of slotted wings proposed by Handley Page. A large monoplane wing with a single long slot has also short slots in front of each aileron. Two other new Handley Page designs have appeared. Earlier trials showed remarkable results. A D.H.9 airplane fitted with a Puma engine was found, when equipped with a Handley Page wing, to have both greater aerodynamic efficiency and greater climbing capacity than with the standard wing. The climbing slope in the one case was 1 in 7.2, and in the other 1 in 10.4. This is important when clearing obstacles around an airdrome. For getting off and onto decks, it was found that the launching run with the Handley Page wing was less than half that necessary with the standard wing under similar conditions.

The ability to fly from or land on the deck of a ship is becoming increasingly important. Amphibians as well as airplanes should be capable of doing so. Descents onto water cannot always be avoided and "landings" of this kind with airplanes are liable to be expensive. Experiments are being made to determine the sea-keeping abilities of the large type N-4 multi-engine flying boats. These weigh about 30,000 lb. and the power developed by its engines totals 2600 hp.

It is difficult to know what to say of the helicopter. Efforts are proceeding and we shall, I hope, be rewarded for our foresight. Mr. Brennan's helicopter has flown to the extent of lifting the pilot and 250 lb. of useful load, an encouraging preliminary flight.

### Gasoline Piping

Perhaps the most troublesome material in present-day aircraft is the rubber tubing used for conveying gasoline. It has to be of "gasoline-resisting" quality. The "petroflex" piping, devised by Blaisdell, affords a very useful means of avoiding both the rapid aging of the rubber and the stiffness of the copper. Petroflex is composed of about 10 layers, glued together, of the intestines of Chinese hogs. Around these layers are layers of canvas, fireproofed. Outside these in turn, as a final protection, is a spiral of wire, which in the case of land machines is of aluminum. This piping appears to be totally unaffected by gasoline, although care has to be taken to keep water away from the inside. Experiments have shown that the piping will stand an internal pressure of 200 lb. per sq. in.

# ETCHING REAGENTS FOR ALLOY-STEELS

SEVERAL series of "sequence etchings" have been tried out at the Bureau of Standards on a specimen of high-speed steel that was in the "as received from the mill" condition. Sequence etching means the etching with two or three reagents in successive order without any repolishing of the section between the etchings, and the taking of micrographs after each etching at the same spot in the microsection to note any changes in the microstructure produced by the last etching as compared with that developed by the preceding etching reagent. The reagents used in various combinations were: (*a*) Dilute $NH_4OH$ solution together with a weak electric current, the specimen being made positive pole; (*b*) a 2-per cent alcoholic solution of nitric acid; (*c*) boiling sodium picrate; and (*d*) Murakami's reagent, which is a solution of potassium ferricyanide and sodium hydroxide at boiling temperature. The results, as judged from the behavior toward the various etching reagents tried, appear to show that there are at least three different constituents present among the imbedded globules or particles in the specimen of high-speed steel, in the "as received from the mill condition," though any attempt to state the nature of these constituents would, at present, involve speculation.

To obtain tungsten carbide in sufficient bulk for making various etching tests thereon, plans are now under way to try carburizing metallic tungsten, ½-in. diameter bar stock, by the "cementation" method.

# The Crankcase Oil Dilution Problem and Its Solution

By WILLIAM F. PARISH[1]

DETROIT SECTION PAPER

*Illustrated with* CHARTS AND PHOTOGRAPH

STARTING with the premise that while the present automotive type of internal-combustion engine was designed to use the volatile fuels, the increased production of these engines has made it necessary to raise the end-point of the fuel with each succeeding year, the author points out that this practice has resulted in the use of a fuel that is not completely vaporized and leaks into the crankcase in a liquid state and dilutes the lubricating oil. The effect upon the viscosity of the lubricant of this fuel dilution is commented on and the viscosity limits of lubricating oils are established. The relation between the viscosity of the lubricant and the efficiency of the engine in which it is used as determined experimentally by C. W. Stratford several years ago is brought out graphically and also in tabular form. How this dilution has forced the refiners specializing in internal-combustion engine lubricants to increase the body of their various grades of oil in response to the demands of the motoring public, the viscosity having increased in some cases almost 95 per cent between the years 1913 and 1921, is pointed out. Tables are presented to substantiate this claim and also to show the differences that existed in 1920 between the lubrication recommendations of three leading oil companies and the actual sales of the three different classes of their product.

The carbon-forming properties of oils and the effect of heavy oil upon engine efficiency are discussed. Although the dilution of the lubricant should be a minimum with one engine operating at full load on the dynamometer stand, results of engine tests extending over the last 10 years show an increase in such dilution of from 2 to 12 per cent. A series of service tests conducted at Chicago slightly more than a year ago showing a dilution of from 15 to 41 per cent in less than 100 miles of running is mentioned as well as other series of tests on passenger cars, tractors and trucks, in which the percentage of dilution increased with the length of the time run, the viscosity in one case decreasing almost 87 per cent after the engine had run 563 miles in a single month. The effects of the diluted lubricant upon the oil-consumption, friction of the engine and the wear are all emphasized. The history of the development of the lubrication of various forms of prime-mover is traced and commented upon at considerable length.

As a solution of the problem of crankcase dilution the author offers a system for crankcase oil regeneration consisting of four main elements by which the fuel and water dilution are automatically removed and the sediment composed of carbon particles, sand and minute pieces of metal filtered out. The operation of this system, which it is claimed will not interfere with the present lubricating system of the engine and functions equally well with the splash and forced-feed systems is described and illustrated. Charts showing the results obtained from comparative test-runs with and without the oil refining system in use are also presented. These indicate that the use of the system practically eliminates crankcase dilution.

[1] M.S.A.E.—Consulting lubrication engineer, Chicago.

THE present automotive type of internal-combustion engine was designed to use the volatile and dry fuels. The automotive industry has been producing so many of these engines that the oil industry has been unable to keep pace with the fuel demand without cutting further into the crudes, and by various and sundry methods has produced fuels with end-points

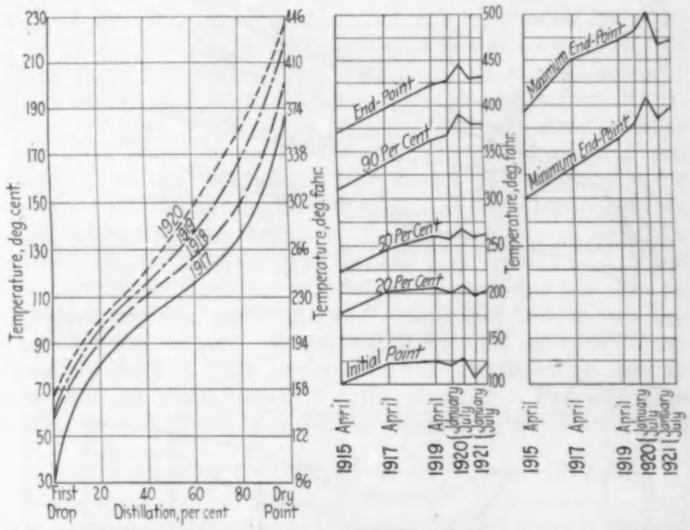


FIG. 1—CHARTS SHOWING HOW THE END-POINT OF THE FUEL SUPPLIED FOR USE IN INTERNAL-COMBUSTION ENGINES HAS RISEN FROM 1915 TO 1921

increasing year by year, as is shown by the charts reproduced in Fig. 1. The result is the unbalanced condition of the original engine, made for dry fuel, apparently using a fuel that passes into the cylinders in a more or less liquid state, where it leaks into the crankcase, combining with the lubricating oil and causing the present serious dilution problem. It is possible to reduce materially the amount of liquid fuel entering the cylinders, by the application of considerable heat and, in this connection, the best results will be secured through using the greatest heat. Heating the mixture to secure better vaporization makes it necessary to lower the compression, which is reflected in lower mileage to the gallon of fuel. It is also possible to catch the flow of raw gasoline in the manifold to some extent, and reatomize it sufficiently to secure better running engines on less fuel. This is being done by many manufacturers and by a number of appliances now on the market. To the extent to which these appliances are used, there is a reduction of the raw gasoline entering the cylinders and draining to the crankcase. Dilution from direct raw-fuel leakages is due to the inability of the carburetion and manifold systems to handle the heavy fuel properly. Any improvements in this direction will naturally tend to alleviate dilution from this cause.

Should the direct leakage of raw gasoline be entirely

overcome, and an acceptable remedy for this condition must be found, there will still be the leakage of the charge in the cylinders during the compression stroke. At this time, while the charge is compressed to a high degree, the mixture is forced by the rings. This mixture contains both the high and low-boiling fractions of the gasoline. Upon entering the crankcase the finely divided oil spray assists the condensation of the fuel, with the result that the lubricating oil in the crankcase absorbs and holds all of this condensed fuel that will not boil away or be redistributed at the engine temperature and condition. As the engine and the oil become colder, the ability of the oil to hold the more volatile fractions increases. If the engine is operated at high temperatures, these volatile fractions will be liberated to a great extent. The fuel leakages are continuous irrespective of the amount of fuel held by the lubricating oil. When dilution is at a comparatively steady point, the total leakage per hour continues, but the engine disposes of the excess.

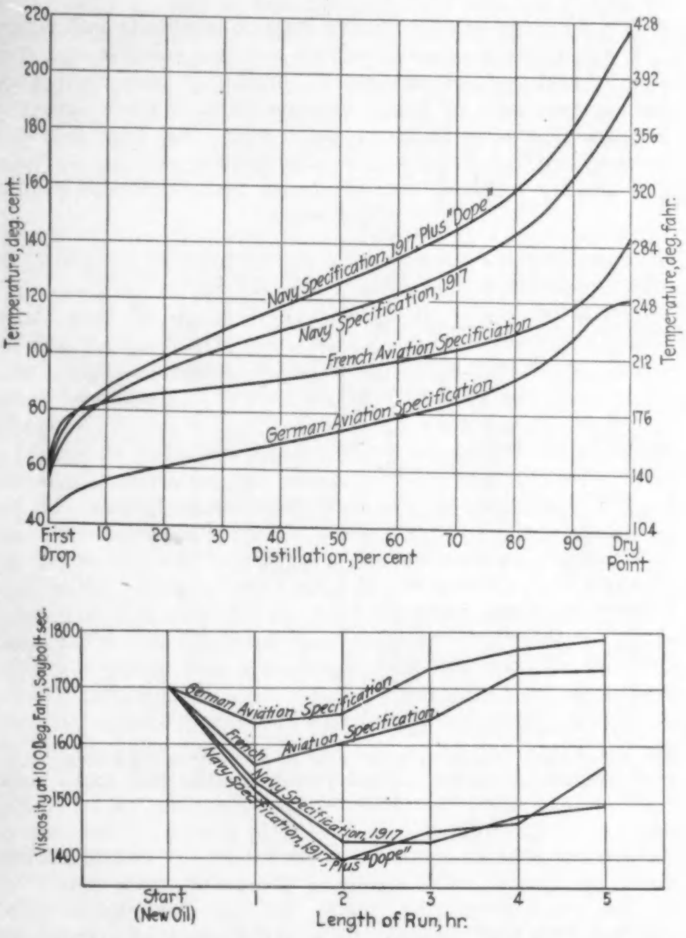


FIG. 2—CHART SHOWING HOW THE VISCOSITY OF A LUBRICATING OIL WAS AFFECTED BY DILUTION WITH FOUR GRADES OF GASOLINE, THE DISTILLATION CURVES OF WHICH ARE SHOWN IN THE UPPER PORTION OF THE ILLUSTRATION

The amount of fuel escaping by the piston-rings during the compression stroke is naturally related to the fuel consumption per brake horsepower-hour. When fuel consumption is low, as a result of fine adjustment and careful operation, dilution of the lubricating oil in a given period of operation is less than when such care is not exercised and the fuel consumption per unit power-output is greater. Dilution is greatest when the operator is unskilful, the rings are worn and the compression and the power output are at a maximum, and intake-manifold vacuum is at a minimum.

TABLE 1 — RELATION BETWEEN THE VISCOSITY OF THE LUBRICANT AND THE ENGINE EFFICIENCY

| Viscosity at 100 Deg. Fahr., Saybolt sec. | Brake Horsepower | Consumption per Horsepower Hour, lb. Fuel | Oil |
|---|---|---|---|
| 100 | 28.0 | 1.40 | 0.0430 |
| 125 | 32.6 | 1.05 | 0.0410 |
| 150 | 33.5 | 0.80 | 0.0380 |
| 175 | 33.7 | 0.70 | 0.0350 |
| 200 | 33.6 | 0.70 | 0.0320 |
| 400 | 33.4 | 0.70 | 0.0270 |
| 600 | 33.2 | 0.70 | 0.0254 |
| 800 | 32.8 | 0.71 | 0.0246 |
| 2,300 | 31.2 | 0.74 | 0.0225 |

The fuel leakage past the piston-rings is naturally dependent upon the condition of the pistons, cylinders and rings. Correctly designed and well-fitted rings, in true cylinders, resist these leakages until a point is reached where the dust and dirt that enters the engine with the carbureter air, in conjunction with the diluted and reduced lubricating oil film on the cylinder walls, brings about a cylinder wall and ring face wear, the worst effect of which is the gradual widening of the piston-ring groove caused by the up-and-down slapping of the ring. This creates a direct passage for the leaking fuel and the escaping or "pumping" oil.

The dilution of the lubricating oil in a given engine was shown in the fuel tests conducted at Washington[2] where four grades of gasoline of the various distillation ranges shown in the upper part of Fig. 2 were used. The lower portion illustrates the effect of the various fuels on the viscosity of the same lubricating oil.

### VISCOSITY LIMITS OF OIL FOR LUBRICATION

The practice of lubrication engineering is based largely on experience. Certain definite rules have been laid down as the result of the experiences of many men who have worked on lubrication problems in all parts of the world. These rules establish limits in viscosity for various classes of work. For instance, the lightest and most rapidly running cotton spindle is lubricated with oils of between 54 and 70 viscosity Saybolt sec. at 100 deg. fahr. Oils lower than the minimum limit cause wear and the blackening of the oil in the spindle base, and that establishes the limit for thinness. Heavier oil than the maximum causes heating of the spindle and loss of power and speed; thus the limit of thickness is obtained. Between these two points the lubrication of the spindle is perfect,

TABLE 2 — CHANGE IN VISCOSITY OF LUBRICATING OILS SOLD BY 10 COMPANIES FROM 1913 TO 1921

| | Light | Medium | Heavy |
|---|---|---|---|
| Average 1913 Viscosity at 100 Deg. Fahr., Saybolt sec. | 162.7 | 209.7 | 250.0 |
| Minimum 1913 Viscosity at 100 Deg. Fahr., Saybolt sec. | 132.0 | 156.0 | 210.0 |
| Maximum 1913 Viscosity at 100 Deg. Fahr., Saybolt sec. | 180.0 | 338.0 | 295.0 |
| Average 1917 Viscosity at 100 Deg. Fahr., Saybolt sec. | 167.5 | 259.4 | 392.0 |
| Minimum 1917 Viscosity at 100 Deg. Fahr., Saybolt sec. | 137.0 | 217.0 | 242.0 |
| Maximum 1917 Viscosity at 100 Deg. Fahr., Saybolt sec. | 205.0 | 330.0 | 868.0 |
| Average 1921 Viscosity at 100 Deg. Fahr., Saybolt sec. | 211.0 | 299.1 | 485.7 |
| Minimum 1921 Viscosity at 100 Deg. Fahr., Saybolt sec. | 176.0 | 196.0 | 226.0 |
| Maximum 1921 Viscosity at 100 Deg. Fahr., Saybolt sec. | 351.0 | 565.0 | 785.0 |

[2] See *Mechanical Engineering*, March, 1920, p. 164.

but if these limits are exceeded detrimental action results. It is to be noted in this connection that the spindle will continue to turn and put twist in the yarn even though it may be undergoing excessive wear or consuming power exorbitantly.

In exceeding the limits of good lubrication practice, the changes that generally take place are too slight and too gradual to be noted immediately. Accumulated, they become of the greatest concern. The experienced lubrication engineer can define limits for the proper lubrication of every classification of machinery. In the case of the internal-combustion engine, however, the lowering volatility of the motor fuels during the last 10 years has introduced several variable factors which make it virtually impossible to standardize the limits of proper lubrication.

When these engines were operated on fuels that caused but little dilution of the lubricants, the oils in common usage in this country were highly finished light-engine or dynamo oils. These oils had an average viscosity of about 190 sec. at 100 deg. fahr.

TABLE 3—RELATION BETWEEN THE LUBRICATION RECOMMENDATIONS OF THREE LEADING OIL COMPANIES AND THEIR ACTUAL SALES IN 1920

| Company | Grade of Oil | Chart Recommendations, per cent | Actual Sales, per cent |
|---|---|---|---|
| *A* | Light | 43 | 20 |
| | Medium | 45 | 60 |
| | All Heavy Grades | 12 | 20 |
| *B* | Light | 5 | 10 |
| | Medium | 85 | 57 |
| | All Heavy Grades | 10 | 33 |
| *C* | Light | 8 | 5 |
| | Medium | 86 | 49 |
| | All Heavy Grades | 6 | 46 |

Fig. 3, which is reproduced from a paper by C. W. Stratford entitled Automobile Lubrication,[3] shows graphically that an oil having a viscosity of 180 sec. at 100 deg. fahr. is the turning point for engine efficiency. Oils lighter than 180 sec. reduce the brake horsepower by increasing the frictional horsepower. The fuel and oil-consumption per brake horsepower-hour are also greatly increased. As the oils used are increased in viscosity, up to 2300 sec. at 100 deg. fahr. the brake horsepower decreases with a slight increase in the fuel-consumption and a reduction in the oil-consumption. The tests from which the results given in Table 1 were obtained were conducted in a testing laboratory owned by an oil company.

### Influence of Dilution Upon the Oil Trade

An extremely interesting phase of the engine oil situation has been created by the dilution problem. Oil companies specializing for this trade have been forced by public demand to increase the body of the various grades of their oils gradually. The motoring public have apparently been the judges as to what their engines required, as evidence will be presented showing that quantities of the various grades of oils actually purchased have not balanced with the charts of recommendations that represent the seller's idea of what should be sold. The oil companies enjoying the bulk of the automotive oil business have seadily increased the viscosity of their brands. The data given in Table 2 bear this out.

[3] See TRANSACTIONS, vol. 10, part 2, p. 86.

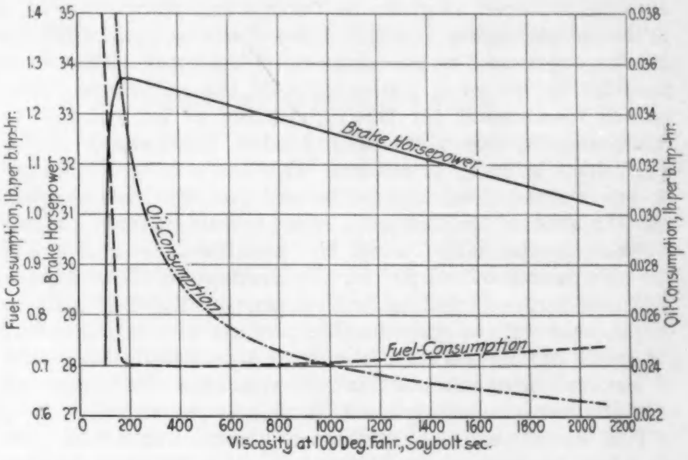


FIG. 3—EFFECT OF CHANGES IN VISCOSITY ON THE BRAKE HORSEPOWER AND FUEL AND OIL-CONSUMPTION OF AN INTERNAL-COMBUSTION ENGINE

The tendency is toward engine oils in general increasing in viscosity. While some of the lighter-bodied oils persist, it has been found that they, as a rule, are not being marketed by companies enjoying the widest trade. The greatest increase in viscosity for all three grades has been made in the best-known trademarked oils. An analysis of the recommendation charts published by three leading oil companies having national distribution shows the interesting tendency of the heavy-oils consumption. The total number of cars, trucks and tractors supposed to be in operation during 1920, as determined for all production models, were allowed a fair amount of oil per car for the season. It was assumed that all cars in operation during 1920 would use the grade of oil recommended by each respective oil company. The distribution of the three grades, in percentages of the total

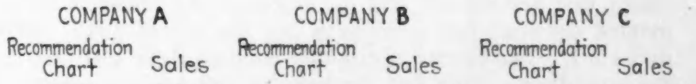


FIG. 4—RELATION BETWEEN THE RECOMMENDATIONS OF THREE OIL COMPANIES FOR AUTOMOTIVE LUBRICATION AND THE ACTUAL SALES OF THE VARIOUS GRADES

amount required, is given in Table 3 and shown in Fig. 4 with each company's actual sales for the year 1920 by grades expressed as percentages of the total. Thus it is possible to see what the experts of the various oil companies recommend for the lubrication of all automotive equipment in the country, and what the owners of the equipment actually purchase. The oils sold by Company *A* are refined from asphaltic and paraffin-base crudes; those marketed by Company *B* are obtained from asphaltic-base crudes only; while the paraffin-base crudes are the sole source of supply for the products of Company *C*.

These figures, dealing with many millions of gallons of oil, and with a considerable portion of the motor-car oil trade of the country, clearly indicate that the public in general demands heavier oils than the oil companies officially recommend in their published charts.

The oldest of the recommendation charts has not changed materially in respect to the grades recommended, since it was originally issued. The oils furnished under the original grade designations have, instead, been changed in viscosity. For instance, one company recommended the same brand and grade for Ford cars in 1913 as in 1921, but the viscosity of this oil has been changed from 170 Saybolt sec. at 100 deg. fahr. to 289 sec.; from 94 to 130 Saybolt sec. at 130 deg. fahr. and 43 to 52 Saybolt sec. at 210 deg. fahr. In accordance with popular grading, this oil was a light oil in 1913, but is now put out as a medium oil. The lubrication of the Ford car has not changed in principle. The engine and the lubricating system are substantially identical with those of the earlier model, and no condition has been introduced that would require a heavier oil, except the dilution, which now occurs but did not exist to a noticeable extent when the lighter oil was supplied.

The unprecedented and unlooked for demand on the part of the public for heavier engine oils greatly upset the oil market for the heavier stocks from which the heavy oils are made. Fig. 5 indicates the trend of automotive demand for lubricants for the years 1919, 1920 and 1921. The prices of the oils used as light oils and as compounding agents for the heavier oils are also plotted. The bright filtered stock, the most expensive of the stock oils, increased very rapidly in price as the extraordinary demands were made upon it. This shortage of the bright stock oil was reflected in the rapid increase in the sale and advance in price of the cheaper filtered cylinder stocks, and finally of the dark unfiltered stocks, which were substituted by a number of manufacturers for the brighter oils with the result that their finished engine oils were black. It is a matter of trade knowledge that every national marketer of automotive lubricants was unprepared for the high and overbalanced demands made by the public for heavy engine-oils during the 1920 season. Figs. 4 and 5 show the way in which the demand developed with the result on the heavy-stock oil market. The advertisements of at least one large oil company during 1921 gave many arguments for not using heavier oil, and can be taken as evidence that there was an attempt being made to counteract the tendency of the public to purchase heavier oils.

### Carbon-Forming Properties of Oils

It is possible to make a laboratory predetermination of the carbon-forming nature of an oil by the Conradson Method, which has been officially adopted by the American Society for Testing Materials for this purpose. It has been determined that the Conradson carbon increases in all oils of the same base as they increase in viscosity. The report of Committee on Standardization of Petroleum Specifications for the United States Government[4] specifies the figures given in Table 4 as the maximum permissible carbon-content by the Conradson method.

TABLE 4 — MAXIMUM PERMISSIBLE CARBON-CONTENT IN OILS SUPPLIED THE GOVERNMENT

| Grade | Viscosity at 100 Deg. Fahr., Saybolt sec. | Conradson Carbon, per cent |
|---|---|---|
| Extra Light | 140 to 160 | 0.1 |
| Light | 175 to 210 | 0.2 |
| Medium | 275 to 310 | 0.3 |
| Heavy | 370 to 410 | 0.4 |
| Extra Heavy | 470 to 520 | 0.6 |
| Liberty Aero[5] | 90 to 100 | 1.5 |
| Liberty Aero[6] | 125 to 135 | 2.0 |

[5] Viscosity values are at 212 deg. fahr.
[6] Summer grade. Viscosity values are at 212 deg. fahr.

Engine tests with various oils indicate that the amount of carbon found on the pistons and in the engine corresponds very closely with the relative amount of carbon of the same oils shown in the laboratory with the Conradson residue test. It is further a matter of common knowledge that the use of heavy oils leaves the engines in worse condition as to carbon. The engines now have to be cleaned of carbon deposits at least every 4000 to 6000 miles, when formerly it was unusual to have this done under 15,000 miles.

Other difficulties encountered while using the heavier grades of oil are lowered mechanical efficiency and difficult starting, especially in cold weather, with heavy loads on the starting-motor causing rapid discharge and short life to the storage batteries.

In general, the oil companies' resistance to the sale and use of heavy oils is based on sound engineering principles. To use one of the heaviest oils in a given engine would increase the frictional horsepower as much as 40 per cent, as compared to a light oil, and would also cause an increase of at least 50 per cent in the frictional temperatures of the bearings, considering the frictional temperature as the difference between bearing and atmospheric temperatures as is brought out in Fig. 6. C. F. Kettering in a paper entitled Fuel Research Developments says that the engine frictions are excessive, and that engine friction is one of the biggest problems confronting the automotive engineer.[7]

Lubrication engineers agree that it is desirable to use the lightest oil that will consistently keep the surfaces apart at working temperatures. The success of internal-combustion engine lubrication with lighter oils was established some years ago, the limits being determined by careful work at a time when there was practically no dilution. With the advent of the dilution problem it was no longer possible to work within such fine limits. In starting in with the correct oil, a high percentage of dilution is produced in a short time. Using a heavier oil merely delays arriving at the same point of thinness. There is an interesting, though complicated, situation in this part of the problem; the engine operates at maximum efficiency with a light oil; the oil industry desires to sell the lighter oil; but the public demands heavier and heavier oils.

### The Dilution of Oil in the Present Automotive Engine

Dilution should be at a minimum when an engine is operating on the dynamometer stand under full load,

[4] See Bureau of Mines Bulletin No. 5; also The Journal, March, 1921, p. 220.
[7] See The Journal, November, 1921, p. 344.

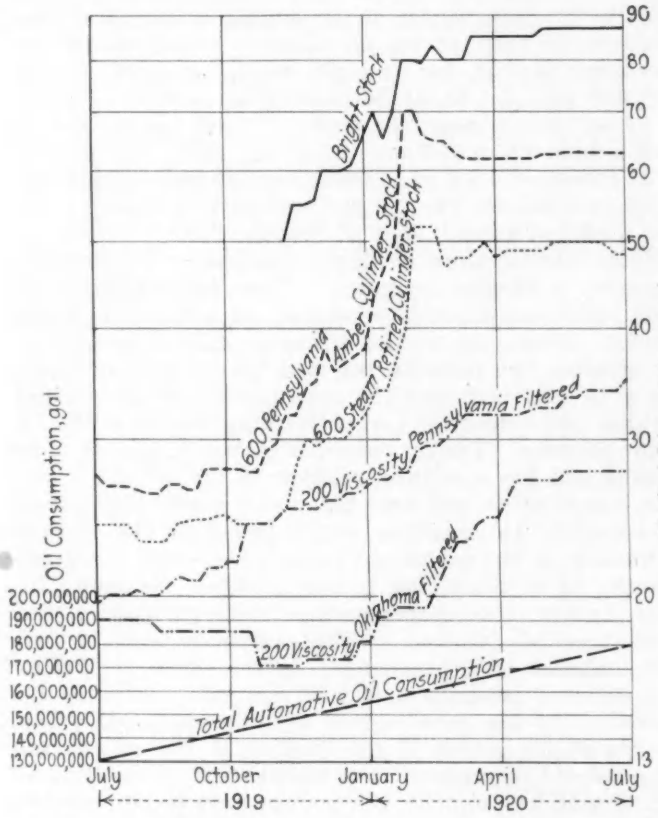


FIG. 5—CHART GIVING THE PRICES OF VARIOUS GRADES OF LUBRICATING OILS FOR THE YEAR ENDED JUNE 30, 1920

with the temperature and fuel controlled and without the effects of starting and stopping. The results of several block-tests of engines under full load conducted intermittently during the last 10 years, are listed in Table 5.

TABLE 5—RESULTS OF BLOCK-TESTS OF ENGINES

| Year | Number of Cylinders | Number of Oils | Number of Refiners | Dilution, per cent |
|---|---|---|---|---|
| 1911 | 4 | 7 | 6 | 2.0 to 9.0 |
| 1913 | 4 | 14 | 10 | 4.0 to 14.0 |
| 1919 | 6 | 7 | 5 | 5.5 to 9.0 |
| 1920 | 6 | 9 | 4 | 7.1 to 12.0 |

In all of the tests mentioned above, the engines were thoroughly cleaned out, flushed with the oil that they used in the test and run from 4 to 10 hr. on the block under prescribed control conditions. The oil was then removed and tested in every particular.

Tests of tractor engines on the block using gasoline of the grade sold in 1919 show dilution of from 8 to 15 per cent. Kerosene-burning tractor engines show dilution of from 30 to 50 per cent in 5 to 8 hr. using 1921 fuel. The question of dilution of the lubricating oil in tractors presents a far more serious problem than that of the passenger car or motor truck. This is because tractors are subjected to a more severe service and the burning of kerosene and the heavier distillates brings in additional complications.

However, from observations in various tests it may be considered that dilution of lubricating oil in tractor engines operating on gasoline in cold weather amounts to between 20 and 50 per cent in several days. In the hottest weather this dilution ranges between 10 and 30 per cent. With kerosene, the dilution runs between 30 and 70 per cent over a period of several days' operation, the per cent of dilution depending upon the original amount of oil in the crankcase, the highest percentages being obtained, of course, in the coldest weather.

## SERVICE TESTS OF DILUTION

An examination of the dilution conditions in nine trucks and cars conducted in Chicago in the winter of 1920 revealed the fact that after runs of under 100 miles the fuel in the lubricating oil amounted to from 15 to 41 per cent and that much of this dilution, in some cases as high as 22 per cent, occurred the first day.

The following winter a special series of tests was made on the engines of cars and trucks equipped with either carbureter or manifold heating devices. One of these, a truck engine that was run the equivalent of 2 miles without going out of the garage, showed 1.5-per cent dilution. Later in service the distance was increased to 50 miles with a resulting dilution of 20.0 per cent and after a further 29.6 miles the dilution increased to 26.5 per cent. An air-cooled engine, run 2 weeks, showed 18-per cent dilution, while a roadster, run 524 miles in 1 week, showed 22-per cent dilution. A passenger car showed 20-per cent dilution after having run 576 miles in 3 weeks, while another of the same make and model showed a dilution of its lubricating oil of 31 per cent after having gone 668 miles in 3 weeks.

A medium-priced car, run 300 miles in 3 days, showed 14 per cent; another, after making 231 miles in 10 days showed 10-per cent dilution and a cheap car's lubricating oil was diluted 11 per cent after making 404 miles in 1 week. A sleeve-valve engine that operated 563 miles in 1 month, registered 47-per cent dilution.

In this last-named instance, the lubricating oil at the outset had a viscosity at 100 deg. fahr. of 360 Saybolt universal sec., while at the end of the test run the viscosity of the oil taken out of the engine was 46 sec.


FIG. 6—CHART SHOWING THE RELATION BETWEEN THE VISCOSITY OF VARIOUS OILS AND THE FRICTIONAL HORSEPOWER AND FRICTIONAL HEAT

at 100 deg. fahr., demonstrating clearly the result of its mixture with the fuel. At a temperature of 212 deg. fahr. which would remove some of the diluent in the sample this mixture still had a viscosity of 36 sec. as compared with a viscosity of 54 sec. for the original oil at the same temperature.

After a run of 214 miles in five days a high-priced car showed a dilution of 22 per cent, while a low-priced car, after operating 380 miles in 4 days showed dilution amounting to 26 per cent. A medium-priced car that had run 445 miles in 1 month showed 27.5 per cent and a high-priced car that had covered 700 miles in 3 weeks revealed dilution of 9.5 per cent.

In a series of experiments with fuels having end-points averaging 436 deg. fahr. conducted between Nov. 12 and Dec. 20, 1920, the data on dilution presented in Table 6 were obtained. In this test, a truck engine showed dilution of its lubricating oil amounting to as high as 46 per cent. Other trucks belonging to the same fleet using the same fuel and the same oil over the same period of days, gave the following results:

TABLE 6—RESULTS OF DILUTION TESTS ON TRUCK ENGINES

| Make | Tonnage | Dilution, per cent |
|---|---|---|
| G | 5 | 17 |
| N | 2 | 13 |
| G | 3½ | 17½ |
| I | 2 | 25½ |
| G | 5 | 14 |
| G | 1½ | 12½ |
| M | 7½ | 11 |
| N | 2 | 31 |
| G | 3½ | 20 |
| G | 3½ | 11 |

### Mass Figures on Dilution

In discussing a paper on Dilution of Crankcase Oil at a meeting of the American Society of Lubrication Engineers in October, 1921, an engineer who is engaged in the installation of plants for the reclamation of used engine oils, reported[8] that in 1921 from collections of 5000 gal. of oil per week in Kansas City he distilled 28.5 per cent of a fraction lighter than light engine-oil. The runs brought about 13.5 per cent of distillate from 55 deg. Baumé to 45 deg. and 15 per cent of distillate from 46 to 38 deg. Baumé.

Another reclamation project at Toledo reported that from more than 500,000 gal. of oil collected from garages and service stations in Chicago, New York City, Detroit and Toledo during the summer of 1921, the company distilled 30 per cent of fractions lighter than light engine-oils. The runs averaged 10 per cent of naphtha of about 52 deg. Baumé gravity, and 20 per cent of distillate of from 45 to 46 deg. Baumé.

### The Viscosity of Diluted Oils

The effect on the viscosity of various oils by the mixture of fuel is shown clearly in Fig. 7. The diluent removed from the lubricating oil after it has run in the engine resembles kerosene more than it does the original gasoline. This is illustrated in Fig. 8. The curves also show that the diluent has a range of boiling points from 100 to 150 deg. fahr. higher than the same fractions of the gasoline that was used for fuel in the engine.

Previous investigations have determined that as a rule the oil-consumption in an engine is related to the viscosity or body of the oil itself.[9] A Curtiss 100-hp. aeronautic engine, for example, has a consumption rate of 0.880 gal. per hr. of lubricating oil with a viscosity of 54 sec. at 212 deg. fahr. and 0.194 gal. per hr. for oil with a viscosity of 135 sec. at 212 deg. fahr. The lighter oil is consumed 4.5 times faster than the heavier oil. Dilution in this case ran 2.0 per cent with both oils. The fuel used had an end-point of 350 deg. fahr.

When dilution thins down the lubricating oil, the consumption naturally increases. This feature must be given due consideration whenever an attempt is made to offset dilution by the use of heavier oils. The heavier oils produce the more carbon and the dilution of these oils with fuel increases the consumption of the oil and yet does not reduce the carbon-forming nature of the oil in the mixture. The formation of carbon is a continuous process and has a definite relation to the amount of oil that creeps up to and over the piston-heads where it is decomposed. In recording engine test data, the amount of dilution in the oil should be deducted from the gross quantity of oil remaining in the crankcase, as shown by C. M. Larson in his paper entitled Determination of the Percentage of Dilution.[10] The remainder will be the actual amount of oil consumed. In the years when dilution was not recognized, it was customary to make no correction in the consumption of lubricating oil. The records of the extent of the dilution in the oil are incomplete and inaccurate. For instance, if at the expiration of a 10-hr. run, the dilution amounts to 20 per cent and the same amount of "oil" is removed as was originally put in the engine, it has been customary to report "no consumption." As a matter of fact, one-fifth of the original oil has been consumed and its place taken up by condensate from the leaking fuel.

### Effect of Diluted Oil on Engine Friction

Approximately one-quarter of the engine friction is due to the bearings and the auxiliaries, the other three-quarters being caused by the piston-rings rubbing on the cylinders. Reducing the viscosity of a heavy oil by mixing kerosene with it will reduce the power required to overcome the friction in the main bearings, until the reduction in the viscosity reaches a point where the surfaces strike. Then there is a rapid increase in frictional power as the striking area becomes larger. Reducing the viscosity of a heavy oil with kerosene for the lubrication of the cylinders will not bring about a reduction of the frictional horsepower to the same degree as with the lubrication of the bearings because the heat of the cylinder-walls will throw off some of the diluent at that point; consequently the diluted oil on the cylinder will be thicker and produce more resistance than the same oil in the crankcase.

My records show that used oil taken from automotive engines with a viscosity of 200 sec. at 100 deg. fahr. scarcely ever appears. Most of the oils taken from the engines are under 150 sec., many are under 100 sec. and a large number are in the 40 and 50-sec. viscosity class, all readings being in Saybolt seconds, at 100 deg. fahr. With any comprehensive standard of proper lubrication for the automotive engine in mind, the automotive or lubrication engineer will readily appreciate that with these viscosities the low limit for safety has been passed and that wear is taking place to such an extent that it will rapidly become detrimental to the engine.

Oil, primarily, is for the purpose of keeping surfaces apart so that they will not strike. If the oil becomes so thin that the film interposed between the two surfaces

[8] See *Scientific Lubrication and Liquid Fuel*, November, 1921, p. 11.

[9] See *Journal of the American Society of Naval Engineers*, vol. 32, p. 45.

[10] See *Scientific Lubrication and Liquid Fuel*, June, 1921, p. 16.

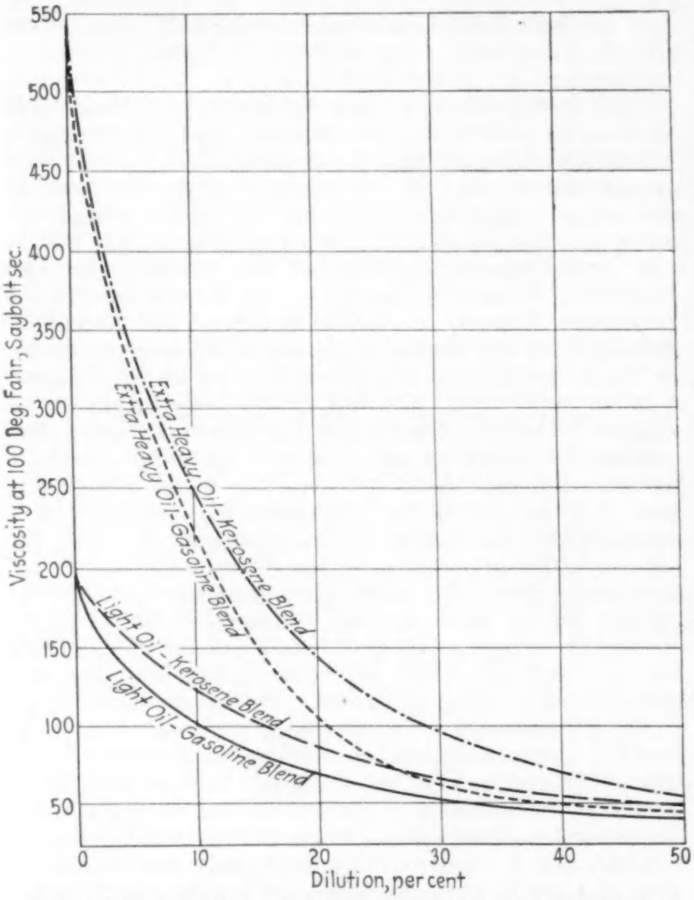


Fig. 7—Effect on the Viscosity of Various Oils of Adding Fuel to Them

is insufficient to prevent contact of those surfaces, local heating is immediately established where the striking takes place and the oil at that spot becomes suddenly thinner at the very moment when thicker oil is needed. As the oil becomes thinner through heat or dilution the locality where the striking occurs becomes more extensive in area and the wear is increased. In the automobile engine this wear may continue for some time as in the case of the wear of the cotton spindle, but eventually it causes a marked deterioration in the physique of the engine. Many of the ills of the engine can be traced directly to this wear. Among the common automobile engine ailments caused by this condition are oval cylinders caused by wear, worn piston-rings, pumping of oil into the explosion chambers, loss of compression, the noisy operation of the engine and burning out of the bearings. In the last analysis engines show such great depreciation largely because of the lack of lubricating properties in the mixtures they are forced to use as lubricants.

### The Development of the Lubrication of Prime-Movers

It is of extreme importance to know the problems of lubrication that have influenced the development of all prime-movers. In general problems of lubrication are the determining factors as to whether a type of machine shall exist or not. A machine that cannot be lubricated will not survive. The history of this development shows a long lane, strewn with the skeletons of discarded engines, once popular, but now obsolete, largely due to lubrication troubles. Types of machines that it has been possible to lubricate have survived and have been of the greatest use to our civilization. Types have improved to stages where they give no trouble and develop into prime necessities.

The Westinghouse crankcase steam engine, popular some 30 years ago, was lubricated by a mixture of water and oil placed in the crankcase. Due to leakages of condensation into the crankcase lubricant and also to leakages or pumping of oil by the piston-rings, the original mixtures were never constant; consequently much difficulty was experienced through the lubricating mixture "livering" up, with a consequent failure of the lubrication, burning out of bearings and repairs due to these causes. Eventually the type was discontinued.

The small high-speed engine of many makes, used extensively before the general introduction of the turbine, was lubricated with oil put into the crankcase. Water leakages caused serious emulsion troubles and many improvements were made in the type before these difficulties were overcome. The engines were never free from trouble from a lubrication standpoint until the oil was run out of the crankcase and through filters and separators. The Willans and similarly lubricated English types all experienced troubles due to the dilution of the lubricant by water and the salts carried by the various waters. The types that have prevailed are those in which provision was made to drain the oil from the crankcase into an outside system of filters, settling tanks, heaters and coolers as conditions required.

All modern steam powerplants, all hydroelectric plants and all of the large gas-engine powerplants throughout the world, have adopted as standard, the principle of treating the oil outside of the engine. This treatment includes removing the dirt and the water, cooling the oil

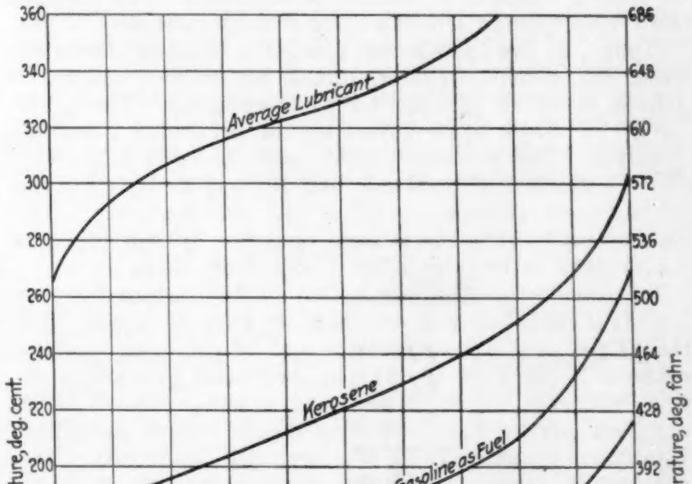


Fig. 8—Distillation Curves of the Average Lubricating Oil, Kerosene, the Diluent Removed from the Lubricating Oil and Gasoline

by water or air or large-capacity storage tanks, and thus securing lubrication with clean, cool oil from which all contamination has been removed.

The development of the steam turbine was marked with about the same cycle of events, but with more disastrous effect, than has transpired in the automobile engine so far as lubrication and dilution are concerned. The first turbines had small reservoirs in the base from which the lubricating oil was circulated fiercely in small quantities. This oil was subjected to changes in temperature and the diluent was water. The general effect was to thicken the oil. The pipes stopped up, the bearings burned out, the blades engaged and the entire machine, in many instances, was wrecked. At the present time, however, as shown by the installation drawings of a modern steam turbine, especially of the marine type, the turbine itself occupies an inconspicuous part, while the drawing is devoted mainly to the details of the lubricating system with its coolers, filters, settling tanks, strainers, pumps, traps and other refinements. Oils have not changed radically but turbine lubrication methods have and now oil is given a better chance to perform the services for which it was intended. As a result, the turbine today is one of the most efficient of power generators and does its work with a minimum of upkeep and depreciation.

The development of lubrication of the Diesel engine was along almost similar lines. In America the first Diesel engines were lubricated by the splash crankcase system. Water and oil were employed. The heat of the engine evaporated the water and the mixture then became too rich in oil with bad "livery" emulsions which, of course, failed to lubricate the bearings. The original American type of Diesel engine is now extinct.

Today, in the installation plan of a modern Diesel engine, the coolers, heaters, filters, separation tanks and pumps make an installation by themselves. The lubrication of the original Diesel engine presented a serious problem. The modern Diesel operates most effectively on any number of good oils and the engine itself is one of the most efficient of all of the prime-movers. The cost of its lubrication is small and the upkeep of parts which have to be lubricated is extremely little.

In considering the lubrication of the automotive engine it is interesting to note that we have developed from the 81 per cent splash systems and 19 per cent pressure systems in the cars of 1911 to the better proportion, in the 1920 cars, of 43.52 per cent splash-pressure, 33.33 per cent pressure and 23.15 per cent splash systems.[11] However, fundamentally we are still lubricating the automobile engine, as in the original designs, with oil placed in the crankcase. In these systems, the oil circulates within the engine and carbon from the burned fuel and oil with dust and dirt from the road are retained by the oil. Particularly is this true of tractors. There are many instances in which tractor piston-rings and engine bearings have been destroyed in a few days due to dust acquired by the lubricant. In addition to this dust and dirt there are also the iron and metal particles, arising from the wear that has taken place. All of the foreign substances help to break-down the oil and wear out the bearing surfaces. The mechanical development of the lubrication of the automotive type of internal-combustion engine has not gone far beyond the original stages. In the meantime the fuel has changed radically, and the fact has been established that the fuel finds its way into the crankcase and mixes with the oil, thus preventing lubrication.

Prof. C. E. Lucke in a paper entitled Rising Importance of the Oil-Injection Type of Internal-Combustion Engine[12] makes the statement that the present engine is no longer commercial unless the leakages of fuel are stopped and the engine is properly lubricated. With the present engines and fuel there is a direct loss of lubricant due to the more frequent drainages of the thinned crankcase lubricant. These drainings are at least twice as often as they were 5 years ago and this represents millions of gallons of oil per season. There is a secondary loss due to the oil being thinner and working up by the rings at a more rapid rate. The use of the heavier oils in an attempt to reduce these losses has placed an extra first cost on the consumer as these oils are more expensive per gallon to make and to buy. The dilution condition has easily doubled the internal-combustion engine lubricating bill of the Nation without giving an adequate return in better lubricated engines. The use of the heavy oils also increases the gasoline consumption, due to the higher engine friction, and the engines have to be cleaned from the carbon from these heavier oils at more frequent intervals than in the past, two items of increased cost of considerable magnitude.

These losses are of small value as compared to the losses involved in upkeep repairs and adjustments necessitated by the lack of lubrication. Add to the above items the depreciation of equipment that, with the hardest service engines, offers almost the greatest sales resistance, and there are sufficient grounds from the technical, the economical, the operating and the commercial standpoints to demand a considerable change in the one feature necessary to put the engine back on its original plane of excellence as far as lubrication is concerned.

### Solving the Lubrication Problem

The problem of dilution is caused through the mixing together of the heavy hydrocarbons of the lubricant with the higher boiling-point hydrocarbons from the fuel. These hydrocarbons can be separated by heat. In order that the separation can be carried on at temperatures considerably below the ordinary boiling-points of the lighter fractions, vacuum distillation must be used. It has been found that even the petroleum oils constituting the lubricants will distill without breaking down or having decomposition take place.

Another way of reducing the temperature necessary to distill the lighter portions from the heavier is by agitation while boiling, a familiar example being the determination of the flash-point of fuel oil in the Pensky-Martin closed tester, where the oil must be stirred when taking the flash to liberate the most volatile fractions at low temperatures. J. E. O'Neill in a paper entitled Fractional Distillation of Lubricating Oils,[13] states that his experiments have shown that by violently agitating the flask it was found that the light oil distilled off at a much lower temperature than it would have if not agitated. He also points out in the same paper that the introduction of steam or carbon dioxide accomplishes the same result. As a laboratory proposition the removal from the crankcase lubricating oil of all the diluent caused by the use of gasoline as fuel can be accomplished without difficulty. In the case of dilution where kerosene is used for fuel there is an overlapping of the end-point of the diluent and the initial points of the lubricant, and possibly but 90 per cent of the diluent can be removed continuously, which would require all kerosene-

[11] See *Automotive Industries*, Feb. 17, 1921, p. 311.

[12] See *Mechanical Engineering*, October, 1921, p. 653.

[13] See *Journal of the American Society of Naval Engineers*, vol. 28, p. 465.

burning engines to operate with a lubricant containing 10 per cent dilution of a very heavy end of the fuel.

It has been established that used oil, with the diluent removed and the dirt and carbon taken out, is as good a lubricant for the internal-combustion engine as new oil. Evidence has been presented to prove that oil is much better after it has once been used in an internal combustion engine, then cleaned of foreign matter and the diluent removed, than when new.[14] It it known that the use of oil that has passed through the engine, and been cleaned of diluent and dirt, will produce less carbon with each successive recovery. This is an indication that the carbon-forming part of the oil is gradually burned out by the engine, and that the most carbon in the engine is produced by the newest oil.

### Temperature of Oil in the Crankcase

The viscosity of all oils is reduced as the temperature increases. At a temperature of 150 deg. fahr. in the crankcase, the light, medium and heavy oils will all be of different viscosities. If the light oil can then be made to operate at a temperature of 130 deg. fahr., it will have slightly more viscosity than the medium oil at 150 deg. If the temperature can be lowered to 120 deg. the viscosity of the light oil will be equal to the viscosity of the heavy oil at 150 deg. fahr. It is possible, therefore, to have the equivalent of a heavy oil in the engine by using a light oil and controlling the temperature.

The viscosity of a diluted light-oil may be the same as the viscosity of the undiluted light-oil but heated to a high temperature. It is, therefore, necessary, after heating the oil and fuel mixture to remove the diluent, again to reduce the temperature of the clean oil to a point at least the same as the temperature at which it is taken out of the crankcase.

### The Crankcase Oil Vac-Refiner

This system for automatically removing the fuel and water dilution from crankcase oil and filtering out the sediment, composed of carbon, sand, metallic particles and the like has conclusively demonstrated both in the laboratory and when attached to the engines of trucks, tractors and passenger cars, as illustrated in Fig. 9, that it is possible to correct all of the present difficulties in the lubrication of internal-combustion engines. The soundness of its basic principles and method of operation has been confirmed by the work of five different groups of engineers, working on the same problem without each other's knowledge or interchange of ideas, who now have combined their respective data and other information into one organization.

This new system of crankcase oil regeneration consists of four main units, (*a*) the heating element, (*b*) the filter, (*c*) the refiner proper, and (*d*) the cooler. The entire system is extremely simple, light in weight, and occupies about the same space as a vacuum fuel system. It does not interfere with the present lubricating system of the engine. It functions equally well with the splash and forced-feed systems. It can be readily installed on most every type of car, truck, or tractor.

Fig. 10 gives a plan of this system, showing the flow of the diluted oil from the crankcase to the heating element and on to the filter and refiner, from where the oil is discharged to the cooler and back to the crankcase. The force causing the circulation of oil through the system is obtained from the vacuum present in the intake-manifold, which ranges from 2 to 25 in. of mercury, according to the type and condition of engine, revolutions per minute, manifold design and other conditions.

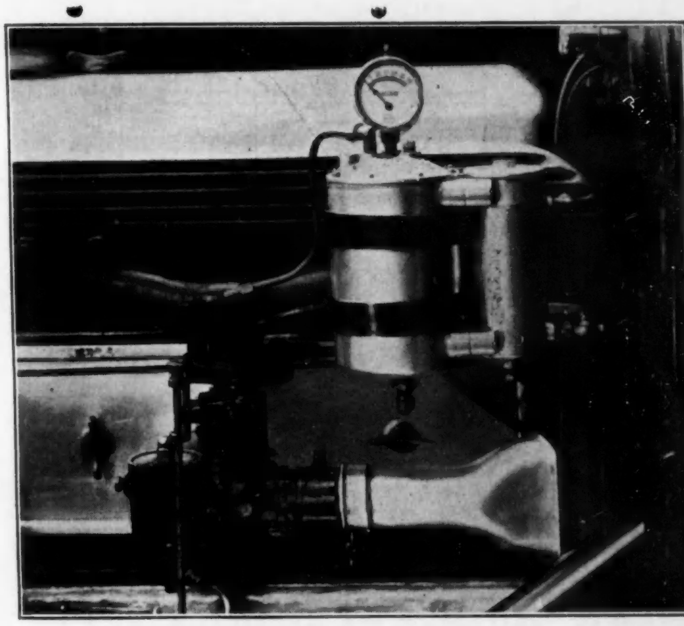

Fig. 9—Application to a Passenger-Car Engine of a System That Has Been Developed for Automatically Refining Crankcase Oil

A tube extends from the top of the refiner to the intake-manifold through which the vacuum, or suction, is transferred to the system, and through which the vaporized diluent is drawn off and burnt in the cylinders as power.

The temperatures shown in the sketch are given as approximately 130 deg. fahr., for the oil from the crankcase to the heater; 400 deg. fahr. from the heater to the refiner and 350 deg. fahr. in the refiner, which temperature plus the vacuum and agitating effect, quickly removes the diluent. From the refiner at 250 deg. fahr. the oil passes to the cooler, with a further falling in temperature in the cooler until the oil will go to the crankcase at about 125 deg. fahr. The diluent, in the form of a fog, or gas, goes into the intake-manifold at tem-

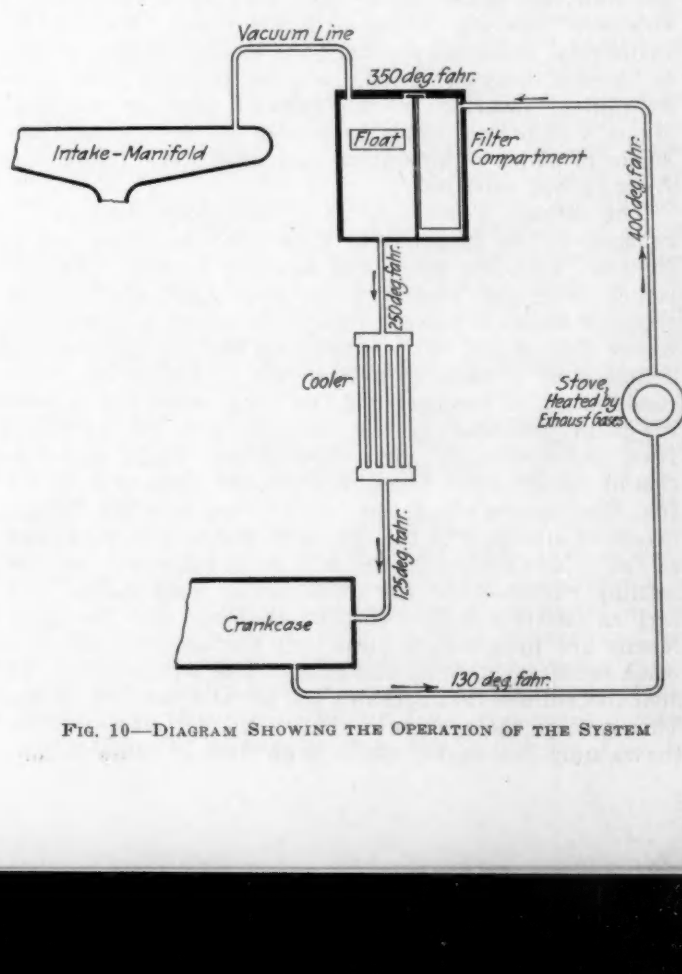


Fig. 10—Diagram Showing the Operation of the System

[14] See *National Petroleum News*, May 21, 1919, p. 20.

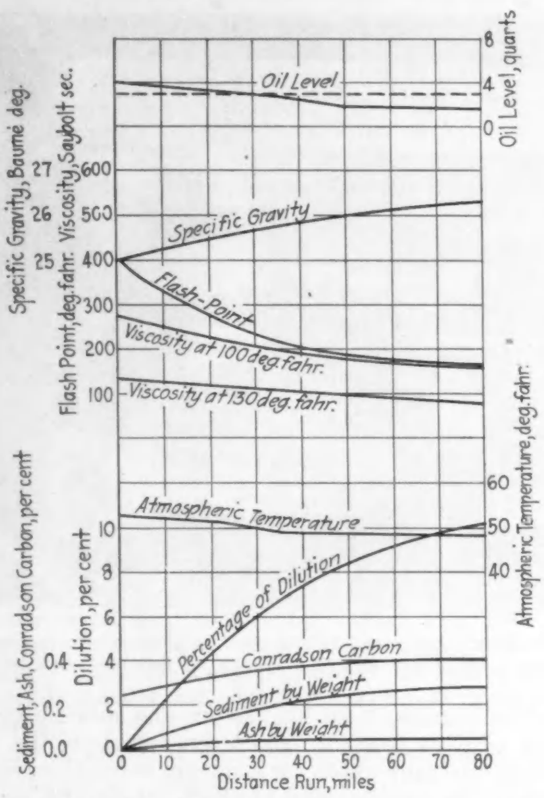


FIG. 11—CHART SHOWING THE CONDITION OF THE OIL IN A PASSENGER-CAR ENGINE THAT WAS NOT EQUIPPED WITH THE AUTOMATIC OIL REFINER

peratures up to 200 deg. fahr., according to the distance of the refiner from the intake-manifold.

Heat is taken from the exhaust by any one of several very efficient ways. There is sufficient heat in the exhaust under practically every condition of operation to allow the removal of most of the diluent from the oil. The heater, while in operation, is either filled with oil to the exclusion of all air, or is working with oil passing through the heater under the force of a vacuum. In this way charring of the oil is prevented. Heaters are cylindrical shells slipped over the exhaust-pipes, or coils of various designs, either machined or made from tubes or piping, inserted in the exhaust pipe or manifold. Heaters of several kinds have been cut open after thousands of miles of operation and have been found free from carbon deposits.

The refiner, which acts as a distillation flask in the removal of the fuel-content from the lubricating oil, is integral with the filter and settling system. The oil comes from the heater to the first filter and settling chamber where it passes through the screen or filter from where the cleaned oil is drawn to the still proper and deflected by a baffle to a thin film of heated oil. The combination of the heat and the great reduction in boiling-point produced by the vacuum, plus the agitation from the moving vehicle, causes a very rapid vaporization of the diluent. The diluent, in the form of a heated fog, then passes along the vacuum line into the intake-manifold and then to the cylinders where it is consumed as fuel. The bottom of the still is arranged as a second settling chamber for the collection of such sludge and dirt as passes the first settling chamber and the filter. Means are provided to clean out the accumulated dirt when necessary, quickly and easily. The still contains the float mechanism that actuates the air and vacuum valves. This part of the system is similar to that employed for the vacuum fuel-tanks, which is an item of value in considering service. There is the one feature of interest as influencing the wear of the only moving parts in the system. With the fuel vacuum-tanks the operating mechanism is mostly dry and occasionally covered with a sulphur powder from the gasoline fumes, while similar parts in this oil refining system are continually covered with oil. The perfectly lubricated parts should, therefore, outlast the engine to which the system is attached.

One of the most important elements in the system is the cooler. The cooler is placed where the air from the fan will dissipate the heat being thrown off from the oil. On engines where there is no fan for cooling, the cooler is placed near the flywheel.

### COMPARATIVE RUNS WITH AND WITHOUT THE REFINER

Fig. 11, which shows the characteristics of many comparative tests, indicates the condition of the oil at various periods in the engine of a car operated in city traffic. The oil has become diluted to an extent of 10.2 per cent in 89 miles. The oil was drained and the engine filled with new oil of the same make and grade, some of the former 10.2 per cent diluted oil remaining in the engine and diluting the new oil about 3 per cent as shown in Fig. 12. After 36 miles the dilution was 1.5 per cent and at 74 miles the dilution was under 1 per cent, where it remained. The viscosity of the oil at the end of the 98-mile run was within 5 sec. viscosity at 100 deg. fahr., of the new oil. The viscosity remained the same in this engine during subsequent runs to a total of 2180 miles. Occasionally during these runs, which were in city traffic, 1 pint of raw gasoline would be put into the crankcase. This would be taken out of the oil inside of 30 miles. One pint of raw kerosene would be taken out of the oil in about 60 miles. The sludge removed from the bottom of the reclaimer tank generally contains a characteristic sediment of 75 per cent of oil, 12 per cent of carbon and 13 per cent of an ash composed of silica and metal. The ash and the carbon will vary with the nature of the roads and the character of the service.

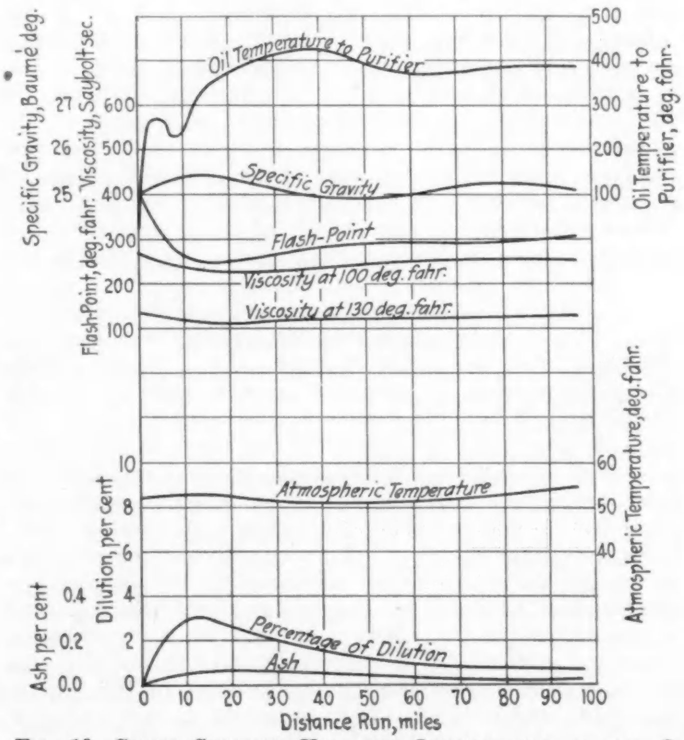


FIG. 12—CHART SHOWING HOW THE INSTALLATION OF THE OIL REFINER MATERIALLY REDUCED THE PERCENTAGE OF CRANKCASE DILUTION

While there will probably be minor difficulties to be overcome in the matter of installation and workout, this system solves the worst angle of the heavy fuel problem, namely, crankcase dilution and its effect upon the lubrication of the engine, and leaves the problem of the proper carburetion or the prevention of flooding of the manifold and cylinders by raw gasoline to be solved at a later date, either through changes in the fuels supplied for use in the future in automotive engines or by the perfection of special mechanical devices.

## THE DISCUSSION

GEORGE L. MCCAIN:—How would you explain the effect of fuel made after the German and French Navy specifications, if after a certain time the viscosity of the oil in the engine were above that at the time the oil was put into the engine?

WILLIAM F. PARISH:—If an engine is run for some time on artificial gas, the viscosity of the engine oil always increases. This is due to the distilling off of the lighter fractions of the oil. In former days when we had a light fuel similar to those made under the above-mentioned specifications, we used light lubricants; the tendency was for the oil to increase slightly in viscosity during use. Where there is no dilution, there is generally an increase in the body of the oil.

GEORGE MEREDITH: — In E. W. Roberts' book[15] it is stated that the claim of dilution due to the incorporating of the lubricating oil with gasoline in the two-cylinder two-cycle engines, is erroneous. What condition would we find?

MR. PARISH:—I do not know where he got his data. He may have found the effect stated in some experiments.

GEORGE A. BREEZE:—What is the temperature loss between the heater and the refiner?

MR. PARISH: — We have had thermometers located right at the heater and at the refiner. There is possibly a loss of 50 or 75 deg. fahr. in the line between those two points.

MR. BREEZE:—Have you made check readings on temperature in the exhaust pipe?

MR. PARISH:—That was covered by O. C. Berry's paper on Manifold Vaporization and Exhaust-Gas Temperatures,[16] in which he shows the range of maximum temperatures in exhaust manifolds. Temperatures are as low as 450 and as high as 1400 deg. fahr.

MR. BREEZE:—What temperature do you find it necessary to maintain to get distillation?

MR. PARISH:—We should have a temperature at the refiner of 350 deg. fahr. when gasoline is used as fuel. To that would be added the effect of the variables, vacuum and agitation, which would be equal to 100 deg. or 150 deg. fahr. There are too many inconsistencies in the practical workout to obtain 100-per cent results. The temperatures and vacuum change as the speeds and loads increase or decrease. Refining the diluted lubricating oil in apparatus on an engine in service is a flexible sort of process that does not work constantly like the speedometers. Of the many variables there is a basic variation or difference in the viscosity of the oil due to the heat in the crankcase. The speed of oil through the system is governed largely by the temperature of oil in the crankcase and this speed causes changes in the oil temperature.

We build up the heat in the oil to 400 or 500 deg. fahr. for kerosene-burning engines. This heat must be liberated in some way before the oil gets back to the engine, and that is done while holding the oil in the cooler during the time the instrument is taking in the fresh charge of diluted oil.

MR. BREEZE:—Do you find that you get your distillation at temperatures less than that of the end-point?

MR. PARISH:—Yes. It frequently is supposed that one must have a temperature at least as high as that of the end-point, but we find that it is not necessary to have so much heat, the vacuum and constant agitation making considerable difference in the problem.

A MEMBER:—Does the refining system have a tendency to increase the viscosity of the oil?

MR. PARISH:—We have not been able to eliminate 100 per cent of the dilution in any of the practical work we have done so far. We think that when we do, the oil will increase in viscosity through the working of the system.

MR. MCCAIN:—If a certain amount of dirt is drawn through the system, does that separate out in the filter and can it be drawn away from there?

MR. PARISH:—Dirt is taken out of the oil in the main filter, which is easily cleaned, and there are two drain-cocks for sludge that passes the filter and settles in the spaces provided; the dirt will be road dust and carbon. If you lubricate properly there should not be much metal in the oil.

L. A. CHAMINADE:—Some of the oils we get at present, such as those made from California crudes and other crudes that have not been properly refined, are emulsifying oils. Due to the products of combustion that pass the pistons and rings, particularly in winter, such an oil soon becomes an emulsion. How does an oil of this type affect this system? How often should the system be cleaned out?

MR. PARISH: — Water in the oil aids the distillation process very materially. The steam bubbling through the oil carries off the light stuff, all going into the engine for power.

MR. MCCAIN: — How often would it be necessary to clean the filter?

MR. PARISH:—Generally, in city driving, one will need to clean the filter every 1000 miles, which can be done very quickly. For tractors in dusty soil, cleaning may have to be done every time oil is put in the engine or water in the air-cleaner. This system holds about 100 cc. of oil (6.1 cu. in.) as a settling space for dirt. Using oil for the collection of dirt, so it can be drained from the two drain-cocks, is a very nice point, because there is always oil containing some dirt that can be taken away from the engine. When there is water in the oil, this aids the distillation and the diluent will distil off about 100 deg. fahr. earlier. In coming off, the water will precede the diluent to the cylinders, being more rapid in travel. With oil containing a compounding material, which emulsifies upon working with water, this refining system would clean the oil and allow the water to go back into the engine cylinders.

MR. BREEZE:—With your system it seems the fuel has slightly different characteristics than one would expect from it. That may be due to the cracking of the fuel in the heater.

MR. PARISH:—In every paper I have read on the subject of dilution it is stated that the dilution comes from three causes: (*a*) the loss of fuel during the compression cycle, (*b*) draining of fuel that has accumulated in the intake manifold and (*c*) some cracking or decomposition of the lubricating oil. In assembling the demonstration apparatus I tested some oil made here in Detroit of 230 Saybolt-sec. viscosity at 100 deg. fahr., running it through the refiner for 30 min. at from 500 to 600 deg. fahr. or

[15] See The Gas Engine Handbook, by E. W. Roberts, p. 236.
[16] See THE JOURNAL, March, 1922, p. 171.

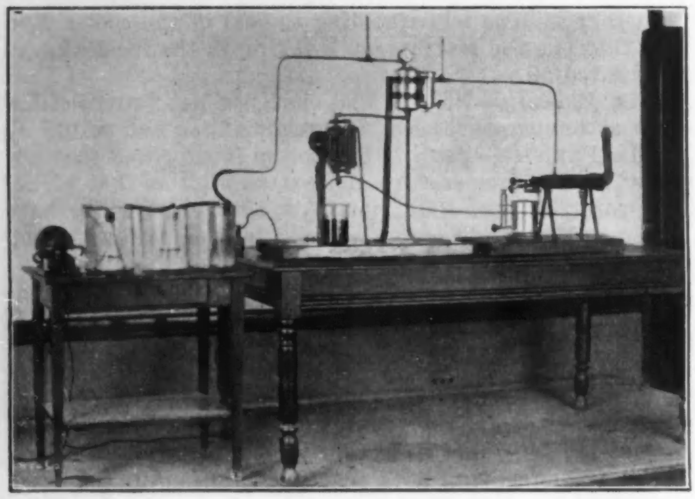

FIG. 13—SET-UP OF APPARATUS TO DEMONSTRATE THE PRINCIPLE ON WHICH THE CRANKCASE OIL VAC-REFINER OPERATES

1000 cc. (61 cu. in.) of new oil with no diluent added, the gravity of the new oil being 23 deg. Baumé. There was a recovery of 1 per cent of clean bright light oil that had a specific gravity of 43 deg. Baumé.

J. R. WADIA:—If under certain conditions in this Country the dilution of oil be somewhere about 10 per cent, will it be more or less in the same engine in India?

MR. PARISH:—It will be much less. The Rangoon and Burma fuels used in India are much more volatile fuels than those we have in this Country. Also, heavier lubricating oil is used on account of the trade and climate. The "light" oil in India and Europe is always as heavy as the "heavy oil" in this Country. It would be an advantage to use a lighter-bodied oil with the more volatile fuels. The conditions are overbalanced in India, the same as they are in England and France. Where there are light fuels and heavy lubricants, the heavy oils last longer, which is the main excuse for their existence. With volatile fuels, the lighter bodied oils should be used for the efficient action of the engine.

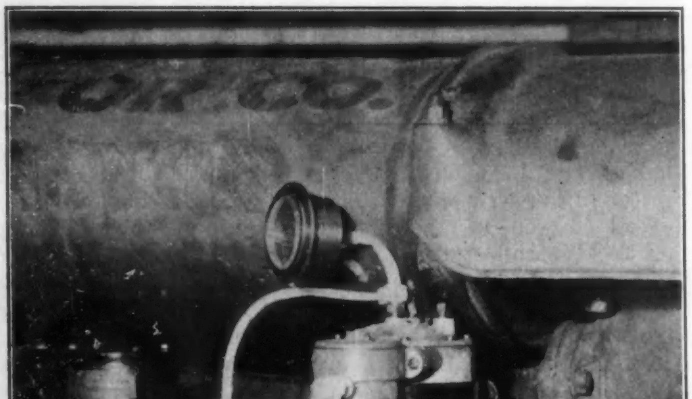

FIG. 14—THE CRANKCASE OIL VAC-REFINER APPLIED TO A TRACTOR ENGINE

MR. WADIA:—Are the oils sold in India under the same brands as the oils sold in this Country the same in quality?

MR. PARISH:—The only way to tell is to analyze the oils. No one on the outside would know without analyzing the oils. As a matter of fact, if a large company were marketing lubricating oil and the brands were well established, there would be no reason against maintaining the brand and meeting the demand for the body by varying it.

G. J. LUX:—How would this refining system handle a compounded oil?

MR. PARISH:—Such an oil might stick up inside the pipe a little, and perhaps the inside of the refiner would look about as bad as the inside of the engine crankcase when using the same compounded oil.

TABLE 7—DISTILLATION TEST OF A GASOLINE-OIL MIXTURE

| Time | Oil Circulation Through Refiner | | | | | Temperature, Deg. Fahr. | | | | Total Oil Shown in "Crankcase" Beaker, cc. |
|---|---|---|---|---|---|---|---|---|---|---|
| | Intake-Manifold Vacuum, In. | Time to Fill, Min. | Time to Fill, Sec. | Time to Drain, Min. | Time to Drain, Sec. | Heat of Refiner | Heat of Vapor | Drain from Cooler | Oil in "Crankcase" Beaker, cc. | |
| 9:55 | 11.00 | .. | 42 | ...... | 9 | 260 | 100 | 110a | 74 | 1,200a |
| 10:00 | 10.50a | .. | 45 | ...... | 10 | 370 | 190 | 120 | 78 | 1,100a |
| 10:05 | 10.25 | 1 | 35 | ...... | 13 | 400 | 240 | 140 | 84 | 1,000 |
| 10:10 | 9.00 | 1 | 40 | ...... | 15 | 490 | 220 | 150 | 90 | 925 |
| 10:15 | 8.00 | 1a | 45a | ...... | 14a | 470 | 208 | 145a | 94a | 900a |
| 10:20 | 12.00 | 1 | 55 | ...... | 11 3/5 | 560 | 210 | 142 | 98 | 875 |
| 10:25 | 9.00 | 1 | 37 | ...... | 16 1/5 | 470 | 265 | 163 | 99a | 750 |
| 10:30 | 7.00 | 2 | 12 | ...... | 12 2/5 | 490 | 250 | 160 | 100 | 750 |

a—Taken from other tests.

K. K. HOAGG:—From a practical standpoint, how often would one need to change the oil in the crankcase, with this refining system in constant operation?

MR. PARISH:—One could run on the same oil all of one season. Why drain the oils when they retain their body and are clean?

L. C. FISK:—Have you any practical figures on the better fuel economy? You are reusing much of the fuel that would otherwise be thrown away. Do you know the percentage of economy you obtain?

MR. PARISH:—The figures I have are not accurate enough to make a statement as to fuel savings. There are many variables and it is not wise at this time to give figures. When carbureters are set for lean mixtures without the refiner, the addition of the refiner makes the mixture rich.

In regard to the demonstration of the Gross crankcase-oil refiner, the apparatus is shown in Fig. 13. The manifold vacuum was represented by using a motor-driven vacuum-pump. The diluent from the mixed oil and fuel was drawn from the refiner through three condensing bottles that were packed in ice. The heated oil was obtained from a type of heating element developed for the use of this refiner system, for the Fordson tractor on which the apparatus is installed as shown in Fig. 14. Heat was supplied from a gasoline blow-torch. A thermometer attached to the refiner showed the temperature of the fuel and oil mixture at that point. Another thermometer was placed in the vapor line to show the tem-

perature of the diluent, or distilled fuel, as it left the refiner and, under engine conditions, would have gone into the engine for fuel. A thermometer was placed also in the graduated beaker, which held the mixed oil and fuel. This beaker represented the crankcase. When the refined oil left the refiner it went through a cooler, back of which was a small electric-motor fan for cooling.

A mixture was made of 750 cc. of Texaco Motor Oil Light and 750 cc. of Red Crown gasoline. This was put into the beaker representing the crankcase. The refiner system required about 300 cc. of the oil to fill the heater, filter, refiner and pipes. After the system drained through the cooler, the number of cubic centimeters in the beaker was measured, the receding amount being caused by the fuel distilling off and being caught in the bottles. Attention was called to the exhaust from the vacuum-pump, which was made up of gas that would not condense. In former experiments this consisted of about 10 per cent of the amount of diluent that had been put into the oil.

The results of the demonstration made at this meeting on a mixture of 50 per cent Texaco motor oil and 50 per cent Standard Oil Red Crown gasoline are shown in Table 7. Upon completion of the demonstration, the diluent in flask No. 1 was 260.0 cc.; that in flask No. 2, 28.5 cc.; and that in flask No. 3, 6.0 cc. The total, 294.5 cc., represents 40 per cent of the original diluent off and condensed in 30 min.

# WHAT GERMANY CAN PAY

WHILE elaborate and apparently convincing arguments can be built upon the present economic facts to prove the contention that Germany will not be able eventually to meet the schedule of payments under the London settlement, an examination of the data upon which all such arguments have been based raises the questions whether on the one hand Germany's permanent losses have not been considerably overestimated and whether on the other hand sufficient consideration has been given to certain intangible, but none the less important, factors in Germany's future productive capacity.

The loss of political control over rich natural resources, upon which other important industries depend, does not necessarily mean the complete loss of economic control. It is true that a change in political control may in part divert the flow of natural resources from their normal channels, but there is much recent evidence that economic forces are overcoming political barriers and it now appears likely that a larger part of the prewar resources of Germany will find their way back to German industries than was at first supposed.

There is also evidence that the economic power of substitution is already working in Germany to supply, from within her boundaries or from neutral countries, materials to take the place of those lost under the treaty of peace. It further appears that insufficient importance has been given to the great possibilities of the development of neighboring territories by Germany and of her ability to establish an even larger German economic unit than existed before the war.

Perhaps most important of all, any estimate of Germany's economic position that considers her saving capacity permanently reduced proportionately to the temporary losses of her natural resources overlooks the fact that one of the most important elements in Germany's productive capacity could not have been permanently reduced by the war, namely, her genius for applying science to business and industry. Indeed, there appears evidence that this genius is not only left intact, but has been increased greatly by the war. Moreover, her extraordinary capacity for organization, which before the war was partially diverted to military operations, can now be directed exclusively to the organization of industry and commerce, not only to the commerce of Germany but also in part to that of the surrounding countries, including Russia.

In reviewing the principal items that condition her future capacity it is possible to believe that Germany will eventually regain an important part of her prewar saving capacity. In any case we are justified in recognizing that there has not yet been produced convincing evidence that Germany will not eventually develop a very great saving capacity.

Such capacity, however, must rest upon the condition that what is left of the German economic unit can become readjusted and operated with an approach to the degree of efficiency that prevailed before the war. There is much to indicate that substantial readjustment is possible provided crushing public financial burdens do not continue to dislocate industry and commerce.

The outstanding aspects of the German problem can be summarized as follows:

(1) The present German problem undoubtedly constitutes one of the largest obstacles to the recovery of trade throughout the world. Indeed, wherever one turns in the maze of European economic and financial problems one is squarely confronted with the German problem

(2) A very large part of the present German problem, so far as it relates to general business conditions and uncertainties in foreign trade, arises directly or indirectly from the deficit financing occasioned by the unbalanced budget

(3) After allowing for all seemingly practicable retrenchment in expenditure and for practical increases in revenue, it appears clear that the balancing of the budget has been impossible in the face of the total of currently accruing charges in the very nature of things, therefore, it is to the interest of the world at large and specifically to the interests of the Allies that there be made whatever temporary arrangements are necessary to enable the budget to be balanced

Once the budget is balanced Germany will be in a position to accomplish the readjustment of her production machine necessary to increase her capacity to make the fullest possible reparations payment.

In the recent tentative readjustment of indemnity payments advancement has been made in the recognition of the real problem. Whether the full temporary relief necessary to permit Germany to recover her maximum paying capacity is to come in the form of a combination of internal and external loans, as a partial moratorium, or through a general readjustment of international obligations is a question upon which much progress is yet to be made. Any acceptable plan must, of course, comprehend the clearly recognizable needs and rights of the Allied Governments, some of which also face tremendous fiscal problems.—*Commerce Monthly.*

# HOT-SPOT METHOD OF HEAVY-FUEL PREPARATION

(*Concluded from page* 32)

carbureter opening and additional contact with the exhaust-manifold a little farther along, usually at a point of division of the fore-and aft reaches. Apparently the second "spot" catches some of the particles that elude the first one and gives a more complete and steady evaporation.

One important requirement of a successful fuel-charge heater is that it should warm-up and get underway quickly. The walls should be thin, and, if cast, should be lightly ribbed on the exhaust side. Aluminum combines low specific heat and rapid conduction and is a very suitable material for a cast hot-spot heating surface, if there are no shielded parts that become heated to such a degree as to melt.

### Preventing Liquid Fuel Reaching the Cylinder

It is recognized generally that it is desirable to prevent liquid fuel from reaching the cylinder and it has been claimed for many designs that they have this action. We have tried models of a number of them and have found that few impede the travel of liquid fuel to the cylinders in even a slight degree. With transversely ribbed elbows, for instance, the fuel drops are caught off the tips of the ribs by the air eddies and snatched through the elbow as if no ribs were present. This, of course, is with air velocities above 70 ft. per sec., and part of the lively action naturally is due to the spheroidal condition already described.

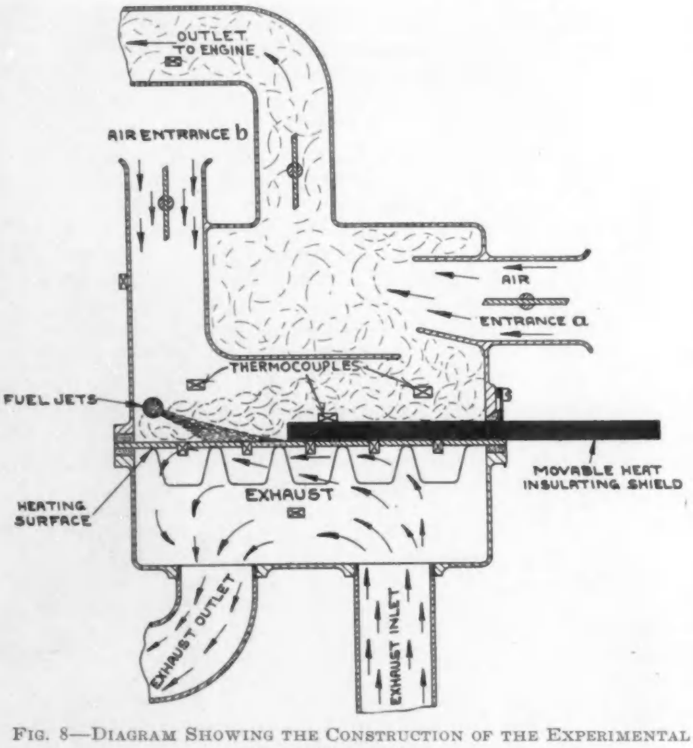


Fig. 8—Diagram Showing the Construction of the Experimental Fuel Heater Used

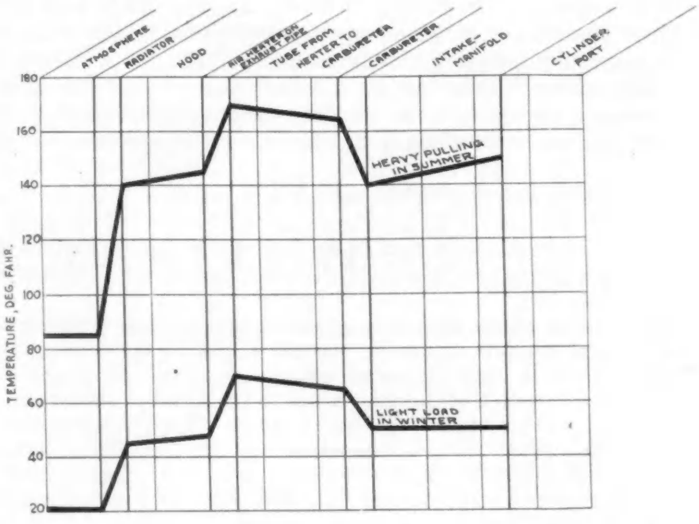


Fig. 9—Curves Showing the Range of the Natural Temperature Variation of the Air Charge

We have used centrifugal force, surface tension and the force of gravity to separate the unvaporized drops. Careful combination of all seems to be required to achieve complete separation. A partial separation, which should be very effective at low engine speeds, can be obtained by abruptly increasing the manifold area above and beyond the hot-spot. This would allow the heavy drops to settle down and again be hurled against the heating surface. The separation and recirculation would obviously be beneficial to the action of either Figs. 4 or 5, but *the heat supply must be adequate or the fuel will not reach the engine*, with an actually functioning liquid-fuel separating device.

In our work with various types of fuel heater, we have experienced a slight but important change in viewpoint, perhaps a keener realization of the truth, in the problem of supplying fuel to internal-combustion engines. This I would like to communicate to the Society. After watching the fuel, in an accumulation equal to many cylinder charges, bubbling, splashing, sometimes lying quiescent on the heating surface of glass-walled hot-spots, and sometimes swept through in a high-velocity spray, one fact stands out: the *rate of fuel-feed from the manifold to the cylinder* primarily governs the conditions of combustion, and the rate of fuel-feed to the manifold is an indirect rather than a direct controlling factor, as regards the mixture proportion of the charge actually used by the engine.

This point of view, we believe, is the proper one from which to consider the problem of the efficient use of fuel in our engines of today.

Vol. XI July, 1922 No. 1

# The Mechanism of Lubrication[1]

*By* ROBERT E. WILSON[2] AND DANIEL P. BARNARD 4TH[3]

ANNUAL MEETING PAPER *Illustrated with* CHARTS AND DIAGRAMS

THE authors state that the coefficient of friction between two rubbing surfaces is influenced by a very large number of variables, the most important being, in the case of an oiled journal, the nature and the shape of the surfaces, their smoothness, the clearance between the journal and the bearing, the viscosity of the oil, the "film-forming" tendency or "oiliness" of the oil, the speed of rubbing, the pressure on the bearing, the method of supplying the lubricant and the temperature. The primary object of the paper is to present the best available data regarding the fundamental mechanism of lubrication so as to afford a basis for predicting the precise effect of these different variables under any specified conditions.

Definitions of the terms used are given and the laws of fluid-film lubrication are discussed, theoretical curves for "ideal" bearings being treated at length. The application of the recommended method of plotting to data in the literature of the subject is described, the thought then including consecutively oiling method and oiliness-factor effects; the effect of variable clearance; bearing metal and clearance effects on the critical point of film rupture; oiliness and miscellaneous effects. A summary is given of the essential conclusions and a description of the method of their application to practical problems.

THERE is no need to discuss or stress the importance of lubrication and a more thorough-going fundamental knowledge of its mechanism before an audience of engineers. The wide discrepancy between the indicated horsepower of the engine and the power delivered to the rear wheels of an automobile calls attention more forcefully to the lack of proper lubrication and a low coefficient of friction than could an entire paper devoted to this phase of the subject.

The mechanism of lubrication admittedly is a very complicated subject. The coefficient of friction between two rubbing surfaces is influenced by a large number of variables whose effects are separable only with difficulty. The most important variables in the case of an oiled journal are the nature and shape of the surfaces, their smoothness, the clearance between the journal and the bearing, the viscosity of the oil, the "film-forming" tendency or "oiliness" of the oil, the speed of rubbing, the pressure on the bearing, the method of supplying the lubricant and the temperature.

It is difficult to separate and determine quantitatively the effect of any one of these variables. For example, in comparing two surfaces it is never possible to be sure how much of the difference is due to the kind of metal used and how much to the degree of smoothness attained in their preparation. In comparing two lubricants, the absence of a definite measure for the film-forming tendency of an oil makes it difficult to know how much of the difference in its behavior is due to variations in viscosity and how much to this rather indefinite oiliness factor. In comparing bearings the variations can be attributed to differences of clearance and of methods of feeding the oil, or to the condition of, and pressure on the surfaces. Similarly, it is very difficult to come to definite conclusions as to the effect of even such simply and readily measurable variables as load and speed. For example, some results seem to show that increasing the load or the speed tends to increase the friction coefficient, while other experiments indicate the opposite. Such discrepancies generally are due primarily to failure to distinguish between the regions of partial lubrication and complete fluid-film or perfect lubrication, the fundamental laws of which are different. If we are to make progress in designing bearings and improving lubricating oils, it is essential to understand the fundamental mechanism of lubrication and to be able to predict the precise effect of the different variables under specified conditions.

## DEFINITIONS

*Speed* is the relative motion of the bearing surfaces in feet per minute. This is obviously equal to $\pi N D/12$, where $N$ is the number of revolutions per minute and $D$ the diameter of the shaft in inches.

*The bearing pressure, p,* is defined as the total load on the bearing (figured vectorially if more than one load is acting in different directions) divided by the projected bearing-area in square inches. In other words, for a cylindrical bearing $p = l/L\ D$, where $l$ equals the total load and $L$ equals the length of the bearing. This is obviously a fictitious value, since the pressure-distribution curve varies considerably in different portions of the bearing, but it serves as a convenient basis for comparison.

The *clearance, c,* as used throughout this paper, is the difference between the diameters of the journal and the bearing. It is therefore twice the radial clearance used by some writers, frequently without clear distinction.

The *frictional resistance, F,* is measured by the amount of force that must be applied in a direction tangent to one of the bearing surfaces to cause it to move past the other surface at the desired speed.

The *coefficient of friction, f,* is equal to this frictional resistance divided by the load; it is that portion of the force pressing the surfaces together that is required to move the surfaces relative to one another. For flat surfaces it can be visualized most clearly as the tangent of the angle at which one loaded surface would just slide down the other at the desired speed under the action of gravity alone. This figure is the best single measure of the efficiency of a given bearing. It does not, however, represent in any sense the ratio between the power transmitted by a journal and that which is dissipated in the bearing.

The *lost power* is the product of the frictional force times the speed. The term "lost work" is sometimes used

[1] This represents in amplified and slightly modified form the first part of the paper presented under this title at the Annual Meeting of the Society, in January 1922. The second part will appear in the August issue of THE JOURNAL under the title The Measurement of the Property of Oiliness.

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loosely in the same connection, but it is generally the lost work per unit of time or the lost power that is really of interest.

The *carrying power* of the film will be discussed more in detail later. For the present it merely will be pointed out that a rotating bearing with an adequate supply of oil tends to drag a fluid film of lubricant between the bearing surfaces. The thickness of this film increases with the speed of rotation and the viscosity of the lubricant but decreases with the load. The carrying power of a bearing is the amount of load that can be carried by a given bearing operating under specified conditions without decreasing the thickness of the fluid film below some definite limit where practical experience shows that there is danger of having actual metal-to-metal contact and abrasion. There is considerable disagreement as to the magnitude of this limiting thickness, which undoubtedly varies with the smoothness of the surfaces.

The *"oiliness"* of a lubricant, for the purpose of this article, is defined as that property by virtue of which one fluid gives lower friction coefficients at low speeds or high loads than another fluid of the same viscosity. It is possessed in varying degrees by different lubricants, ordinary mineral oils generally being somewhat deficient in this respect as compared with most animal and vegetable oils. It is tacitly assumed throughout this paper that the oiliness of a lubricant is connected in some way with the adsorption of some constituent in the oil by the metal surface and this assumption is believed to be justified fully by data to be presented in the subsequent paper on The Measurement of the Property of Oiliness.

### Fundamental Laws of Fluid-Film Lubrication

Since nearly all well designed journal bearings operate under conditions of perfect fluid-film lubrication during the major portion of their operating lives, no consideration of the mechanism of lubrication would be complete without a detailed discussion of the fundamental laws of fluid-film lubrication and their application to practical problems in the design and operation of bearings. As already indicated, the coefficient of friction is a function of a large number of variables and the comparison of results obtained by different experimenters under varying conditions therefore becomes extremely difficult; in fact, it is so difficult that little has been attempted along these lines. The potential value of such a correlation, however, is great enough to justify the attempt.

The best methods of comparing such results will be graphical in nature, rather than involving the application of a complicated equation with the necessarily large number of variables. Certain graphical methods have been employed in foreign publications, particularly those by Lasche.[4] His results are presented in the form of a series of three-dimensional diagrams in which the coefficient of friction is used as the ordinate in all the diagrams and different pairs of the four principal variables, viscosity, speed, load and clearance, are used for the other two coordinates in a given diagram. This requires six three-dimensional diagrams to cover the subject fully, and they are very difficult to construct and use. It seems desirable, therefore, to make use of another and much more convenient expedient.

A recent publication by Wilson, McAdams and Seltzer[4] has shown how the very complicated case of the variations in the friction coefficient $f$ in Fanning's formula with velocity, viscosity, density, pipe diameter and roughness, can be simplified greatly by taking advantage of the fact that $f$ is a factor "without dimensions" and therefore cannot be an *independent* function of the four variables, $v$, $z$, $s$ and $D$, but only of their *combination* in the form $D\ v\ s/z$. By plotting observed values of $f$ against varying values of this ratio, it is possible to bring out very clearly the nature of this functional relationship and the effect of the fifth variable, the degree of roughness, on the values of $f$. Similarly, for the case of lubrication, Hersey[4] has shown by dimensional reasoning that, while $f$, the coefficient of friction, tends to increase approximately in proportion to the speed and the viscosity of the lubricant and in inverse proportion to the load or pressure on the bearing, it is also a factor without dimensions, and hence it cannot be an independent function of these variables, but of a combination of them in the form $z\ N\ D\ L/l$, or its equivalent $z\ N/p$ where

$z$ = Viscosity of the lubricant at the operating temperature
$L$ = Length of the bearing
$D$ = Diameter of the journal
$l$ = Load on the bearing
$N$ = Revolutions per minute
$p$ = Pressure on the bearing

It is, of course, also a function of other factors without dimensions, such as the ratio of the clearance to the diameter, the ratio of the length to the diameter, the smoothness of the bearing, the method of oiling and the like, and possibly of some "oiliness" factor possessed by the lubricant. By plotting the values of $f$, obtained by various investigators under different conditions against $z\ N/p$, it should be possible to tell just how important these other variables are and whether or not they can be ignored within the limits of good bearing design.

In spite of the great potential value of this method for correlating the experimental results of different investigators, it apparently has never been made use of for this purpose. While the results of the dimensional reasoning can be accepted without question, it must be kept in mind that dimensional reasoning specifies only the existence and not the nature of the functional relationship between $f$ and $z\ N/p$.

For most actual bearings with their many disturbing factors of variable clearance, surface roughness, end-leakage and the like, the calculation of the precise functional relationship is difficult if not impossible, and plotting experimental values of $f$ against $z\ N/p$ is the only way to establish the characteristic curve for a given type of bearing. Once this curve is established, however, for a given bearing, dimensional reasoning *does* demand that the same curve represent all other bearings that are geometrically similar in their ratios of clearance and length to diameter, the nature of the surface and the like. It may aid in the interpretation of such curves to first consider quantitatively the behavior of an "ideal" bearing, and qualitatively the probable effect of certain disturbing factors in which all actual bearings differ from the ideal.

### Theoretical Curves for Ideal Bearings

If an ideal bearing be defined as one in which the journal is perfectly centered in the bearing, with absolutely smooth bearing surfaces, no oil-grooves, no end-leakage and the clearance space filled with fluid, a little consideration indicates that, since the frictional resistance is entirely in the fluid film, it is determined solely by the force required to move the molecules of the fluid past one another. In accordance with Poseiulles' law for viscous flow, this force must be directly proportional to the

[4] See bibliography at end of paper.

speed of motion and to the viscosity of the lubricant and independent of the pressure.[5]

The coefficient of friction would therefore vary inversely as the pressure, and if the values of $f$ were plotted against $z\,N/p$, they should all lie on a straight line passing through the origin as indicated by the full lines in Fig. 1. It also is obvious that the friction will be greater the smaller the clearance is. Indeed, it is readily possible to calculate the equation for the ideal line, using merely the laws of fluid friction, such calculation giving $f = 48 \times 10^{-9} \cdot D/c \cdot z\,N/p$, where $c$ is the diametrical clearance. For a typical clearance ratio of 1/1000, $f = 0.000048\,z\,N/p$. The lines drawn in Fig. 1 were calculated for clearance ratios of 1/1000 and 1/250 respectively.

### Actual and Ideal Bearing Differences

In regard to the probable effects of different factors the most important deviation in actual practice from the ideal conditions are that

(1) The journal is never perfectly centered in the bearing

(2) End-leakage is usually an important factor rather than a negligible one

(3) The bearing surfaces are never smooth

(4) Many bearing surfaces are cut up with oil-grooves, although it is now recognized generally that the presence of such grooves on the high-pressure side of the bearing is very undesirable

Sommerfeld[6] has taken the first step in modifying the above ideal equation and has calculated the corrections arising from the first of the above-mentioned non-ideal conditions, the fact that a loaded journal never operates in a perfectly central position. Such a position is, of course, approached fairly closely at high speeds or viscosities and low loads; or in other words at high values of $z\,N/p$. If the journal does not deviate more than 20 per cent from the central position, the increased rate of shear on the high-pressure side is almost counterbalanced by the decreased rate on the other, and the value of $f$ is therefore affected but little. As $z\,N/p$ decreases still further and the eccentricity increases correspondingly, the value of $f$ deviates more and more from that calculated for the central position. The results of Sommerfeld's rather complicated derivations may still, however, be plotted against $z\,N/p$ and the lines thus calculated for values of $c/D$ of 1/1000 and 1/250 are shown dotted in Fig. 1. It will be noted that these lines do not continue indefinitely to approach the origin, but pass

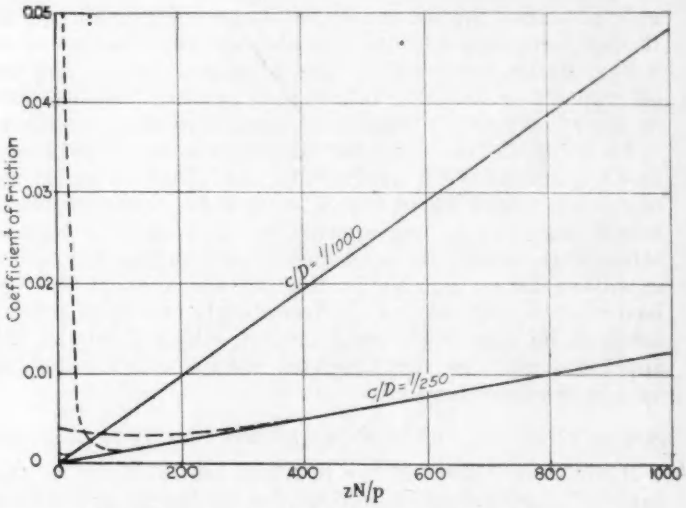


Fig. 1—Predicted Curve for the Ideal Bearing

through a minimum and then rise slightly for further decreases in $z\,N/p$. The magnitude of the deviation and the value of $z\,N/p$ at which it becomes appreciable are both much greater at the higher clearances, as is indicated by the two dotted curves in Fig. 1 that actually cross at low values of $z\,N/p$. It must be emphasized, however, that Sommerfeld's derivations make correction for only one of the four important disturbing influences present in most actual bearings and do not take into account the effects of end-leakage, surface roughness and oil-grooves which may well be of equal or greater importance.

The probable qualitative effects of some of these other disturbing factors can, however, be anticipated. Thus end-leakage undoubtedly will tend to keep the supporting film thinner than would be the case otherwise, especially if the clearance is large or the length of the bearing small compared to its diameter. As a result, bearings with large clearances should behave more like those with smaller clearances and give higher coefficients than those predicted by Sommerfeld. The magnitude of these effects can be estimated only by plotting actual results as is done in the next section of this paper.

The impossibility of getting perfectly smooth bearing surfaces also has a very important effect on the problem. Sommerfeld's calculations indicate a practically constant value of $f$ for low values of $z\,N/p$ clear down to zero; whereas it is common knowledge that, if $z\,N/p$ is decreased too far, by making the appropriate change in any of the three factors, the fluid film is ruptured and the value of $f$ increases enormously. This discrepancy between prediction and observation is recognized by Sommerfeld as arising from his assumption that *any* film, however thin, would keep perfectly smooth metal surfaces apart; while actually, when the thickness decreases to a certain point comparable with the roughness of the surface, metal-to-metal contact and abrasion take place with a sharp rise in the coefficient. Contrary to the implication of several writers, this critical point does *not* bear any real relation to the "point of minimum friction" calculated by Sommerfeld. In practice, the rise in the coefficient is very sharp. In general it is similar to the dashed-line curve drawn in Fig. 1.

The effect of oil-grooves on the high-pressure side of a bearing also would be to thin out the pressure film because of the tendency of oil to flow through the grooves from the high to the low-pressure portion. This effect probably would be less pronounced at high values of

[5] It must not be overlooked that both the pressure and the speed of motion may have an additional indirect effect on the friction by changing the viscosity of the lubricant, and it is essential throughout this discussion to consider the viscosity as that of the lubricating film at the actual pressure and temperature prevailing therein.

As to the effect of pressure, recent published work by Hersey, supplemented by a private communication, and in England, has indicated that very high pressures markedly increase the viscosity of oils, especially mineral oils as contrasted with animal and vegetable oils. These increases are, however, only a matter of 5 to 15 per cent at 40 deg. cent. (104 deg. fahr.) for the maximum nominal pressures of 1500 lb. per sq. in. for which journal bearings are generally designed. For pressures tenfold as great as this, which may be approached in portions of a poorly aligned bearing or in reciprocating parts, the increase in viscosity is about threefold for animal and vegetable oils and sixfold for most mineral oils. Recent work done here by Professor Hersey indicates that these effects are much less marked at higher temperatures.

The indirect effect of speed on the viscosity is due, of course, to the change in the temperature of the oil-film in the bearing and is not so readily calculable; but by installing a properly insulated thermocouple in the bearing, with the junction in direct contact with the oil-film and polished off to conform with the bearing surface, it is possible to determine the working temperature of the film with very satisfactory accuracy, except possibly at very high speeds. The viscosity of the lubricant at any temperature can be determined readily by measuring it at two or three temperatures and plotting the results according to the methods suggested by Dean and Lane, by W. H. Herschel and by Wilson, McAdams and Seltzer.

[6] See bibliography at end of paper.

$z\ N/p$, where the oil could not escape nearly as fast as it was being brought in, and also at very low values of $z\ N/p$, where the pressure film is extremely thin and the oil would find difficulty in escaping rapidly from the film *to* the oil-grooves, unless they were very close together.

In spite of the fundamental importance of Sommerfeld's work and the probability that factors which he neglected might be of equal or greater importance in actual practice, no one appears to have made a serious attempt to check his predicted curves with the rather extensive data available in the literature as to the behavior of actual bearings. Accordingly, we have undertaken to do this in the next section, which treats of the application of the recommended method of plotting to data in the literature.

## Application of the Recommended Plotting Method

It has been shown in the previous section that, on the basis of dimensional reasoning, the coefficient of friction of any given bearing must be *some* function of the modulus $z\ N/p$, rather than an independent function of the separate variables as is generally assumed. In other words, if a series of observed values of $f$ on a given bearing are plotted against $z\ N/p$, they will all approximate a smooth curve, regardless of what combinations of $z$, $N$ and $p$ determine the value of the modulus. The position of the curve thus obtained for a given bearing should indicate the effects of various factors in bearing design, such as clearance, surface roughness, end-leakage and the like. It should afford a logical method of comparing the efficiencies of different bearings even though they may have been tested under widely varying conditions.

A cursory study of the literature appears to indicate that a great wealth of data is available for drawing such plots; but in many cases closer inspection indicates the absence of some one or two essential items of data, generally the temperature-viscosity curve of the lubricant used, or the operating temperature of the bearing. These can and have been approximated from data in the literature in the cases of some well known animal or vegetable oils, but for mineral oils this method fails.

### Oiling Method and Oiliness-Factor Effects

Investigating the effects of the method of oiling and the oiliness factor in the fluid-film region, the most extensive series of tests on any single bearing on which full data are available, where the effects of variable clearance, length and the like do not enter, is in an unpublished thesis at the Massachusetts Institute of Technology by Professor Hersey. The tests were made on a normal full 1 x 3-in. bronze bearing with a clearance ratio of 1/250. Using a variety of speeds and loads, Professor Hersey tested four oils that might be expected to vary widely in oiliness and did vary greatly in viscosity. These were sperm, lard, machine and engine oil. He also used four different rates of oiling: 2 drops per min., 10 drops per min., 30 to 60 drops per min. and bath lubrication. By plotting 400 observed values of $f$ against $z\ N/p$ on a single chart,[7] about 30 x 120 in. in size, and using different colors for the several methods of oiling and a different kind of point for each oil, it was possible to develop many features not capable of clear demonstration within the limits of a printed page. For example, an inspection of the plot indicated very clearly that 90 per cent of the points lay fairly close to a single straight line and that the remaining 10 per cent, which were scattered and higher, were all of a single color and corresponded to the 2-drop-per-min. rate of oil-feed. The different kinds of oil showed no consistent deviations either above or below the line that represented all the points.

Fig. 3 shows *all* Hersey's points for values of $z\ N/p$ below 1100, except the points mentioned above which were obtained when the rate of oil feed was only 2 drops per min. The results with the different oils are differentiated by the type of point used. The remarkable simplification and coordination produced by this method of plotting the data obtained under such a wide variety of conditions are brought out clearly by the fact that a single straight line represents practically every point within the limits of error. These are certain to be fairly large in any measurements of friction. The equation of this line is $f = 0.002 + 0.000018\ z\ N/p$. The slope is higher than that specified by Professor Hersey[8] which, in these units, would be 0.0000164; but it fits the points better, especially those of the engine oil at very high values of $z\ N/p$. Professor Hersey's article contains no plot for comparison.

This degree of agreement would indicate the validity of the following important, though tentative conclusions:

(1) Confirming the conclusions of dimensional reasoning, the method of plotting $f$ versus $z\ N/p$ for a given bearing, method of oiling and the like, gives a consistent series of points approximating a definite line, regardless of what combination of the three variables determined the value of $z\ N/p$. In other words, it does not matter whether the coefficient of friction is measured at a high load and a low speed, or vice versa, provided the resultant value of $z\ N/p$ is the same.

(2) For a given value of $z\ N/p$, the method of oiling, not including the effect of oil-grooves, does not affect the values of $z\ N/p$, providing *enough* oil is supplied to keep the bearing *full*. Lesser amounts give high and scattered results, but anything more, however supplied, has no effect on the fundamental relationship between $f$ and $z\ N/p$.

(3) Within the range studied by Professor Hersey, the oiliness factor of different lubricants had no effect upon the results, the viscosity being the only significant variable

The importance of the last tentative conclusion is so great that the results of a single series of experiments, however carefully carried out, must not be considered as adequate proof. But the conclusion is in entire agreement with the results of Herschel,[8] Holde,[8] and others, according to which even non-lubricants such as glycerine or sucrose solutions gave the same results as the best lubricants in the region of fluid-film lubrication. If oiliness is due to the adsorption of a very thin film of something on the metal surface, this film would scarcely be expected to affect the results appreciably when the surfaces were separated by a relatively thick fluid film in the range covered by Professor Hersey's experiments, however important it might become in the region of partial lubrication where the fluid film has been ruptured.

It should be said, however, that a few isolated experiments, which have been reported, appear to contradict this conclusion, but all these results were obtained on abnormal types of bearings or parts of bearings, where

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[7] Convenient engineering units rather than absolute ones were chosen for all the calculations. Thus $N$ is in revolutions per minute, $p$ is in pounds per square inch of the projected area, and $z$ is in centipoises relative to water at 68 deg. fahr., where it has a viscosity of 1 centipoise = 0.01 poise. In Hersey's experiments, $z$ varied from 38 to 440; $N$, from 280 to 1200; and $p$, from 40 to 255. Fig. 2 gives a conversion chart, modified from Herschel's data by a method first used by MacCoull, for obtaining the viscosity in centipoises from the density and time of efflux of any fluid in any of the standard viscosimeters.

[8] See bibliography at end of paper.

there is always a likelihood that some portion of the bearing, especially the "on" edge, may have been operating under conditions of partial lubrication. Again, as we *approach* the point of film rupture at very low values of $z\ N/p$, some results which will be discussed later indicate that the property of oiliness may begin to manifest itself. There appears to be little doubt, however, as to the lack of significance of the oiliness factor in the major portion of the region of fluid-film lubrication.

In addition to the results shown in Fig. 3, Professor Hersey made a few observations at values of $z\ N/p$ between 1100 and 8000. Most of the high values were obtained by using a very viscous engine-oil. The results have been plotted on a large scale and are rather scattered, some approximating the line fairly closely but others giving considerably higher coefficients, the average deviation being about 20 per cent. This can well be explained on the basis of the observation by Professor Hersey that it was exceedingly difficult to get the very viscous oil to feed-in properly through the small oil-holes. The abnormally high values of $f$ probably are due to the bearing not being entirely filled with oil and are similar to those obtained with the less viscous oils when an insufficient quantity of lubricant was supplied. On the whole, the results with the engine oil confirm, rather than invalidate, the extrapolation of the previously obtained straight line to these very much higher values of $z\ N/p$.

As already indicated, not only does the agreement of all the observed points with a single line permit us to draw important conclusions, but the position and shape of this line is a measure of the efficiency of the bearing, and also affords a method of determining the actual effect of the non-ideal conditions neglected by Sommerfeld in his calculations regarding the behavior of theoretical bearings. Before proceeding to this phase of the discussion, however, it seems desirable to present other data on other bearings for comparison with the results of Hersey and the predictions of Sommerfeld.

### Effect of Variable Clearance

Probably the most important single factor in the design of bearings is the clearance. Practically the only

[9] See bibliography at end of paper.

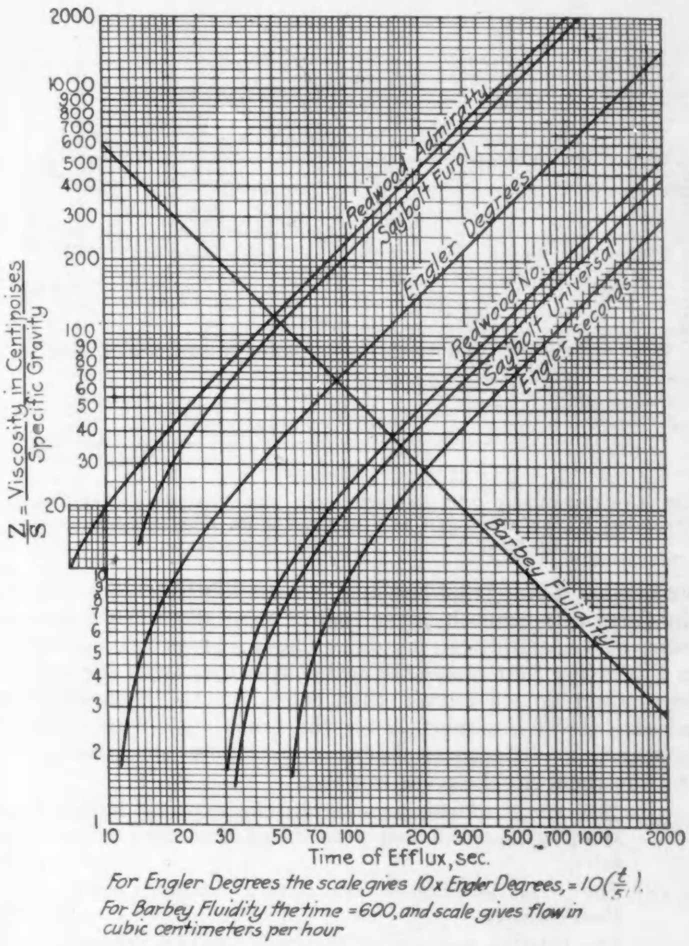


Fig. 2—Conversion Diagram for Viscosimeters

extensive published results in which this is the only variable appear to be those obtained by Lasche on a gun-metal bearing 260 x 110 min. (10.20 x 4.33 in.) against a steel shaft. Lasche's article[9] does not give his original experimental points but expresses his results in the form of a series of surfaces on three-dimensional diagrams in which two factors are kept constant. Using the rec-

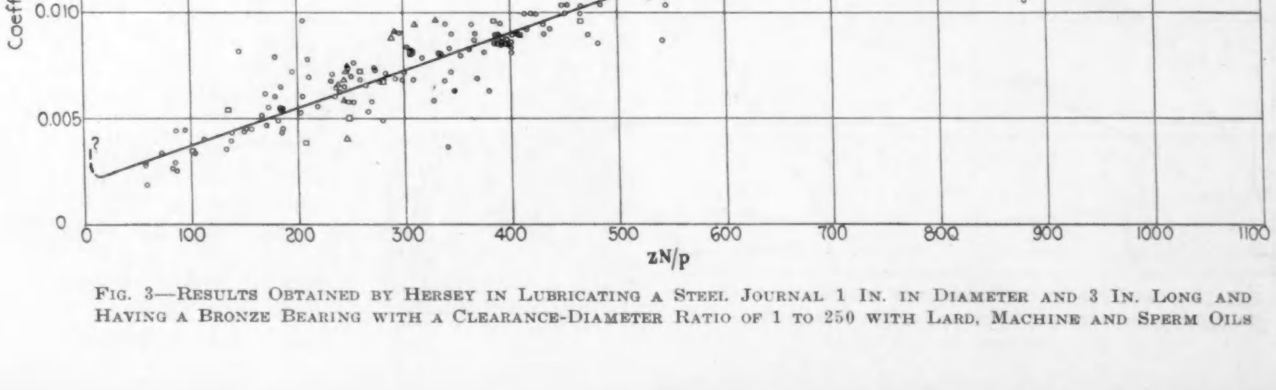


Fig. 3—Results Obtained by Hersey in Lubricating a Steel Journal 1 In. in Diameter and 3 In. Long and Having a Bronze Bearing with a Clearance-Diameter Ratio of 1 to 250 with Lard, Machine and Sperm Oils

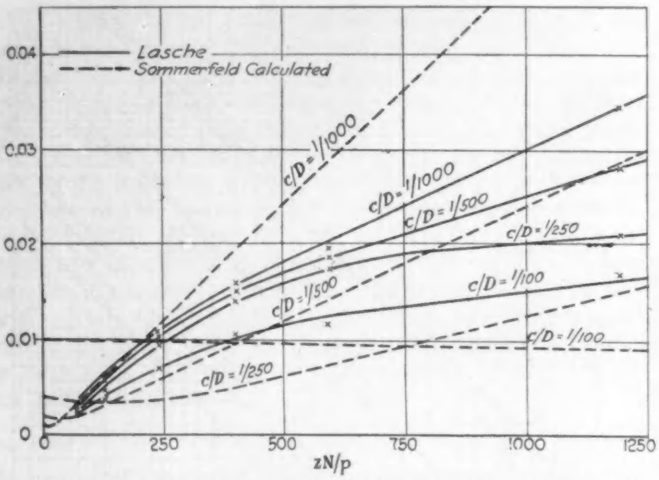


FIG. 4—COMPARISON OF LASCHE'S DATA FOR VARIOUS CLEARANCE-DIAMETER RATIOS OBTAINED AS THE RESULT OF EXPERIMENTS WITH THE CALCULATED RESULTS OF SOMMERFELD

ommended methods of plotting, his results therefore take the form of a smooth curve rather than show the agreement of experimental points. These curves are shown in Fig. 4 for four clearance-ratios, together with the corresponding dotted curves that represent the theoretical calculations of Sommerfeld for the same clearances. The outstanding differences between Lasche's observations and the theoretical curves are that

(1) The effect of clearance is in the expected direction, but is much less pronounced than that calculated
(2) The observed curves are convex upward rather than concave
(3) The relative position of the lines for the different clearances remains unchanged down to the lowest observed values of $z\ N/p$, instead of crossing as Sommerfeld's calculations would indicate

These discrepancies are so striking that Lasche's results could be accepted only with reservation, especially since his surfaces were badly cut-up with oil-grooves, were it not for the fact that the three differences are all confirmed by the rather extensive results of Heimann.[10] Unfortunately, Heimann did not record the viscosities of his lubricants and hence his results cannot be plotted against $z\ N/p$ to determine their positions, but he presents several curves for two different clearances in which the oil and the load were identical and only the speed was varied. The shape and relative positions of the curves $f$ versus $N$ for different clearances were almost identical with those of Lasche in the foregoing three respects, as is illustrated by two typical curves in Fig. 5. Heimann's bearings were ring-oiled and had no oil-grooves on the pressure side. The interesting portion of Heimann's curves in the critical region are discussed in the following section of this paper.

[10] See bibliography at end of paper.

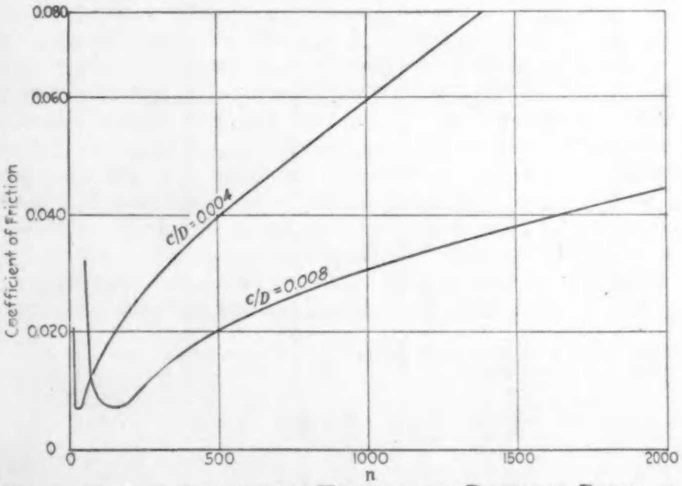


FIG. 5—RESULTS OBTAINED BY HEIMANN FOR DIFFERENT RATIOS OF THE CLEARANCE IN A BEARING TO ITS DIAMETER

The comparison of Hersey's results with those of Lasche for the same clearance indicates a fairly good agreement at high and low values of $z\ N/p$, but considerably higher results in the intermediate region. As indicated in the previous theoretical discussion, this is what might be expected due to the effect of the oil-grooves present in Lasche's bearing.

### FACTORS AFFECTING THE CRITICAL POINT

The results obtained by Stribeck and Heimann are considered in discussing the effect of bearing metal and clearance on the critical point of film rupture. Unfortunately, neither Hersey nor Lasche extended the measurements to values of $z\ N/p$ low enough to reach the point of film rupture and enter the interesting region of partial lubrication where the effects of the oiliness factor and of the nature of the bearing metal should begin to manifest themselves. By far the most extensive results in this region for which all the necessary data are given are those of Stribeck.[10] He worked with two bearings, both of the same diameter, 70 mm. (2.76 in.), but of different lengths, 137 and 230 mm. (5.4 and 9.06 in.), and made of different metals. Data are not given as to the clearance, both having been scraped-in. From the appearance of the curves it would seem that the bronze bearing had the larger clearance and hence gave somewhat lower coefficients over most of the range covered. Stribeck's original data are presented in a long series of families of curves but, again, plotting against $z\ N/p$ all his observed values of $f$ (except a very few on the first run on white metal before the bearing was properly "run-in") produces a remarkable simplification of results, the points for each bearing approximating a smooth curve, as is shown clearly in Figs. 6 and 7, the latter expanding the lower part of the scale.

Fig. 7 is the more interesting in that it shows very clearly the behavior of these two bearings in the critical region and brings out the sharp rise in the coefficient as the value of $z\ N/p$ is lowered below the critical point. In this region it is apparent that a change in the viscosity, the speed or the load has exactly the *opposite* effect on $f$ that the same change had in the region of fluid-film lubrication. This accounts for the apparently contradictory results of a number of investigators who did not distinguish clearly between the two fields. The most important point brought out by Fig. 7 is that for some reason the two bearings differed enormously in their ability to maintain a continuous fluid-film and keep within the realm of perfect lubrication. The curve for the bronze bearing breaks away when $z\ N/p = 13$, while the white-metal bearing kept its fluid film until $z\ N/p = 1$, which is as low as any observations of which we are aware.

The importance of these differences is even greater than might appear from casual inspection. If we take a factor of safety of 5 in calculating the proper value of $z\ N/p$ for given operating conditions, this would permit operation of the white-metal bearing at a $z\ N/p$ value of 5 and a coefficient of friction of 0.0033; whereas, with the bronze bearing, operation with an equal factor of

safety would have to be carried on at a $z\ N/p$ value of 65 and a coefficient of 0.0065.

It becomes extremely important, therefore, to determine the fundamental cause for this marked difference in the behavior of the two bearings. A similar advantage of white metal over bronze was reported by Herschel.[11] It might be due to any one of the following causes, or to a combination of them: (*a*) a smaller clearance in the white-metal bearing that might help to maintain the fluid film longer; (b) a smoother initial surface on the white-metal bearing that permitted the film to thin out farther before rupture; (*c*) the softer character of the white metal that makes it possible to bed-down surface irregularities and prevent highly localized pressures and a consequent rupture of the film; (*d*) a specific adsorptive property of the white metal that might tend to hold a very thin semi-solid film of lubricant and prevent metal-to-metal contact even after the true fluid-film had become extremely thin.

There are two other conceivable causes of the differ-

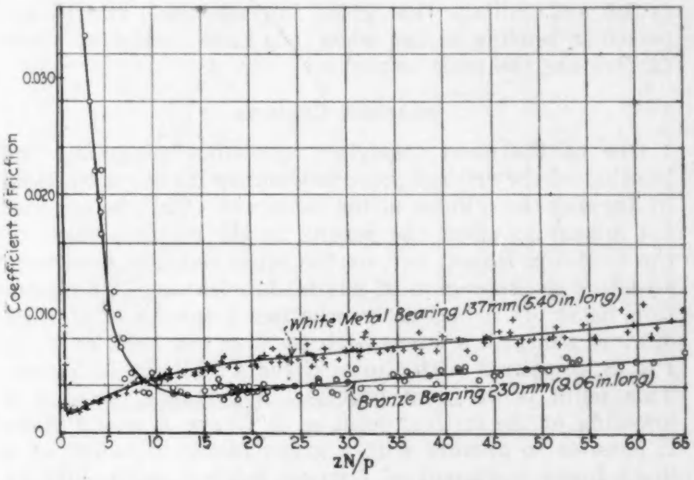


FIG. 7—DATA OBTAINED BY STREIBECK ON BRONZE AND WHITE METAL BEARINGS WITH A STEEL SHAFT 70 MM. (2.76 IN.) IN DIAMETER

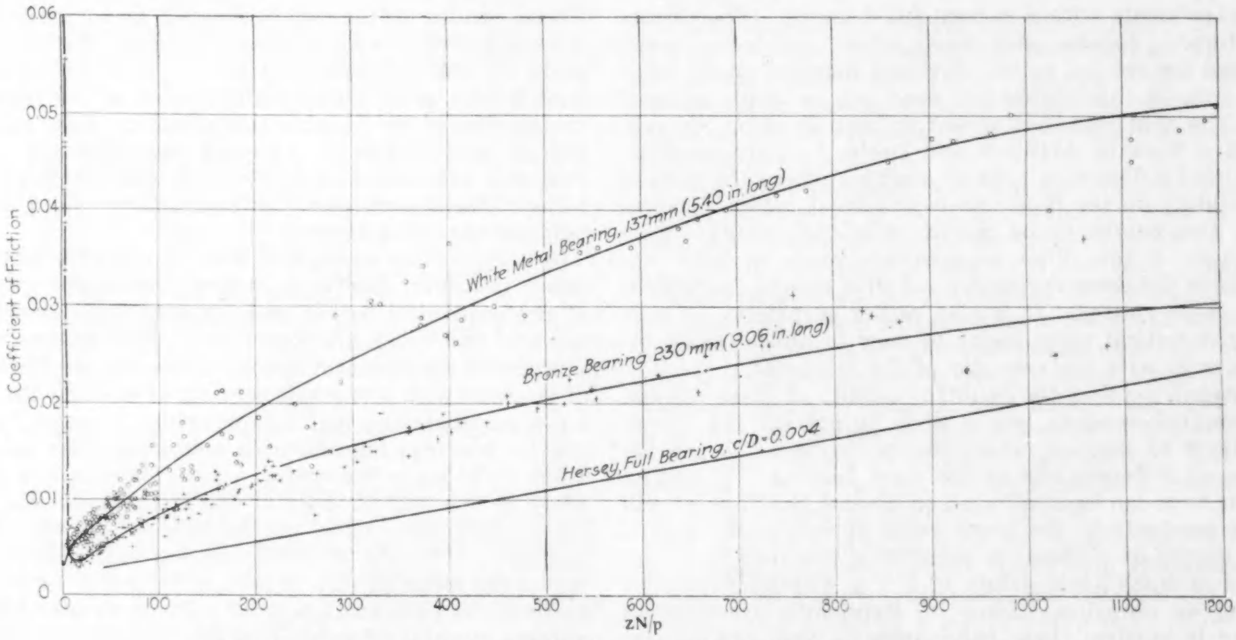


FIG. 6—COMPARISON OF STRIBECK'S RESULTS WITH THOSE OBTAINED BY HERSEY

tioned in the above list: (*a*) length, because the longer bearing would be expected to hold the film better, although actually it held it more poorly; (*b*) the possibility of improper alignment of the bronze bearing, or a similar disturbing factor. This possibility seems to be ence in behavior of the two bearings that are not menruled out because the bronze bearing gives lower coefficients than the white metal for the higher values of $z\ N/p$.

There is evidence in favor of each of the four possibilities above mentioned. The results of Heimann given in Fig. 5 show similar results where the clearance was apparently the only variable; but in Stribeck's case the difference in the clearances does not appear to have been great enough to account for the great lowering of the critical point. Other evidence appears to support some of the other possibilities. To make a definite choice from these four possibilities until more evidence is available would be to speculate unduly. The data presented bring out the importance of comprehensive experimentation with different types of bearing in this critical region of lubri-

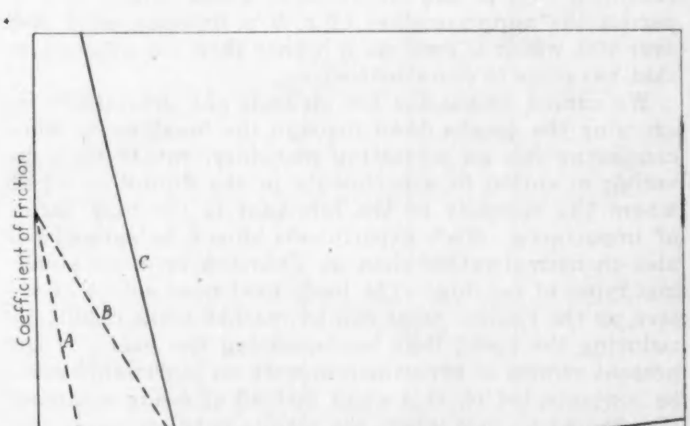


FIG. 8—CURVES SHOWING THE EFFECT OF THE OILINESS OF THE LUBRICANT AT THE CRITICAL POINT

[11] See bibliography at end of paper.

cation and indicate that great improvements can be expected in bearing design when it is known which of these factors are the most important.

### Oiliness Effects

One of the most important questions regarding the location of the critical point is whether it can be changed by varying the oiliness of the lubricant. This factor does not appear to affect the results in the major portion of the fluid-film range; but, on the other hand, it does have an effect in the region of partial lubrication. The question naturally arises as to whether a lubricant of high oiliness will give a curve such as $A$ or one such as $B$, in Fig. 8, compared with Curve $C$ for a less oily lubricant. This point is of great practical importance, because a lowering of the critical point as in Curve $A$ would make it possible to *operate* with a given factor of safety at a much lower coefficient of friction, while a curve such as $B$ could only affect the results under abnormal conditions of very low $z\ N/p$. In spite of its importance no data appear to have been published on the subject based on experiments with a normal full bearing. The results of Hersey, Lasche and most other observers, never reached the critical point. Stribeck obtained many valuable data in this region but used only a single mineral oil. The only data that appear to bear at all on the subject are those of Archbutt and Deeley.[12] Unfortunately, they used a Thurston type of machine where the method of building up the fluid film is abnormal, which detracts from the weight to be placed upon the results. Furthermore, a few other experiments made at very low speeds in the same region did not give results concordant with those plotted. It is also rather surprising to note that the critical point seems to vary in almost exact inverse ratio with the viscosity of the lubricant.

Recognizing then the doubtful validity of these results, it nevertheless seems worth while to present the curves in Fig. 9 to indicate what *may* be found as to the behavior of different oils in the same bearing. If results of this type can be duplicated on normal bearings, it will prove conclusively the great value of lubricants with a high degree of oiliness, in permitting bearings to be operated at much lower values of $z\ N/p$ without danger of seizing or abrasion. Some of Heimann's experiments appear to confirm these conclusions in that the critical value of $z\ N/p$ for some glycerine solutions, such as are shown in Fig. 10 and on which he gives enough data to permit the approximation of $z\ N/p$, is apparently well over 100, which is very much higher than any other value that has come to our attention.

We cannot emphasize too strongly the desirability of carrying the results down through the break point, when comparing oils on oil-testing machines, rather than devoting attention to experiments in the fluid-film region where the viscosity of the lubricant is the only factor of importance. Such experiments should be carried out also on normal rather than on Thurston or other abnormal types of bearing. The loads used need not be excessive, as the critical point can be reached more readily by reducing the speed than by increasing the load. If the present volume of experimental work on lubrication could be concentrated on this point instead of being scattered over the whole field where the results have comparatively little significance, we might confidently expect to add greatly to our knowledge of lubrication within a comparatively short time. It is possible that there are many unpublished data that, if put into the recommended form, would settle some of the very points about which we are now most uncertain.

It must, of course, be recognized that working in this range of partial lubrication will result in abrasion of the surfaces and require their frequent renewal if check results are to be secured. The data obtained by Stribeck indicate, however, that this factor may not be so serious as anticipated if one is careful not to go too far into the region of partial lubrication. In a subsequent article we shall discuss the quantitative *measurement* of this property of oiliness and the methods by which one can investigate the region of partial lubrication much farther than is possible or desirable by using a conventional bearing.

### Miscellaneous Effects

There are a number of other isolated experiments covering more limited ranges that seem worth presenting. The best of these are the results cited by Kingsbury[12] on a light, high-speed bearing with a very small clearance, 1/3750, in which air alone was the lubricant. These results are shown in Fig. 10 in comparison with the calculated results of Sommerfeld for the same clearance. It will be noted that in this case the correspondence is very good and probably is due to the remarkable smoothness of the bearing surfaces and to the minimization of end-leakage by the small clearance and the fact that one end was closed. The normal operating conditions of the Sperry gyrostabilizer are indicated also by a point on the same figure.

A considerable amount of data is available on more or less "imitation" bearings, where the normal conditions of pressure distribution and constant total clearance do not hold and which are therefore of much less value from a practical standpoint. Among these are the Tower type of machine with a one-half bearing or less and the Thurston type where the load is applied by pressing two one-quarter bearings together. In neither of these cases does a film build up in the normal way. There is also a possibility of one part or edge of the bearing section operating under conditions of partial lubrication while the remainder is in the fluid-film region and the resulting curves are meaningless. Again, it has been found experimentally that the coefficient of friction obtained in some of these machines depends to a considerable extent on the precise method of rounding the bearing at the edge where the oil enters. If this is kept sharp the coefficients generally are higher than those obtained under similar conditions for full bearings, while if the edge is ground off in a proper curve to wedge in a thick film of oil it will sometimes give *lower* coefficients than normal full bearings of average clearance.

As a result of all these sources of trouble in these special machines, it is not surprising that many of the data are discordant and frequently do not give a good line when plotted against $z\ N/p$. Some of the best data, however, give fairly definite lines and to make comparisons possible, some results recently obtained by Prof. H. W. Hayward, of the Massachusetts Institute of Technology, on a Thurston-type Riehlé-machine with well rounded entrance edges, are plotted in the customary manner and are shown in Fig. 11. The agreement is obviously very good, in spite of the fact that $N$ varied from 158 to 389, $z$ from 55 to 105, and $p$ from 20 to 500. Furthermore, the whole curve is remarkably low, indicating that the wedging effect of the rounded entrance edges produced a large effective clearance. For comparative purposes, lines representing Hersey's results and the friction experiments by B. Tower[12] are shown in the same illustration.

[12] See bibliography at end of paper.

As noted previously, there are many additional data in the literature that appear to afford further evidence on some of the points as yet unsettled; but an inspection has always indicated that some of the essential data were missing. These are generally either the temperature-viscosity curve for the lubricant or the operating temperature of the bearing. The failure to secure and record such data in any modern investigation is virtually unpardonable.

### Summary of Essential Conclusions

In view of the necessary length of the foregoing detailed discussion, it seems desirable to state briefly the essential conclusions that appear to be justified by the data now available. We appreciate fully the fact that in some cases the data are not really adequate to substantiate these tentative conclusions, but nevertheless it seems desirable to state them in definite terms so that confirmation or contradiction can be offered more readily by other

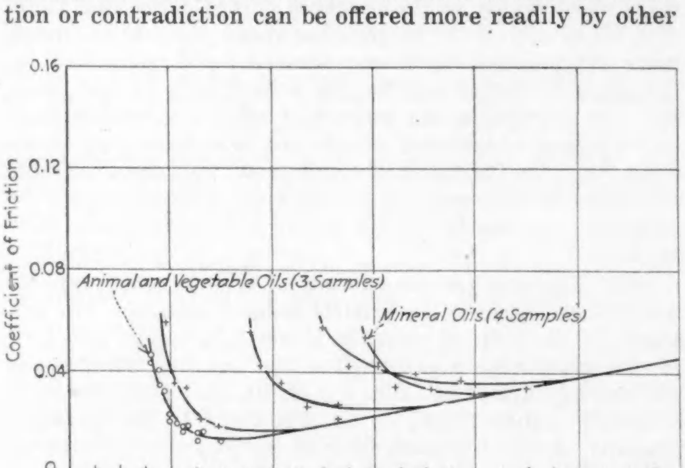


Fig. 9—Results of Lubricating the Same Bearing with Different Oils

investigators, possibly from data already obtained but not published. Certainly such a procedure is preferable to the perpetuation of the present hazy uncertainty and empiricism that pervade most of the literature on the subject. With this object in view, therefore, the following tentative conclusions are submitted.

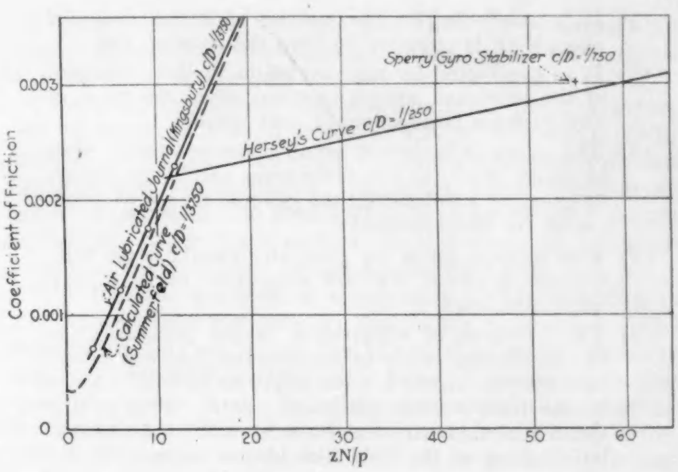


Fig. 10—Comparison of the Results Obtained by Kingsbury with an Air Lubricated Journal with the Predicted Curve for the Ideal Bearing of Sommerfeld and the Results Obtained by Hersey

I—For any normal journal bearing operating in the region of fluid-film lubrication, the coefficient of friction is not an independent function of the speed, the load and the operating viscosity of the fluid at the operating temperature, but is rather a function of their *combination* in the form $z\,N/p$.

II—If the bearing and lubricant are both kept constant, all the observed values of $f$ approximate a smooth curve when plotted against $z\,N/p$. Starting at very high values of $z\,N/p$ and decreasing them, all the curves gradually approach a small value of $f$, generally in the neighborhood of 0.002. Before the ordinate axis is reached, however, the fluid-film is ruptured, the coefficients rise very sharply with a further decrease in the $z\,N/p$ value and metal-to-metal contact and abrasion take place. The first region is termed the region of fluid-film or perfect lubrication; the second, the region of partial lubrication. The intermediate point at which the coefficient is a minimum is the critical point of fluid-film rupture.

III—In the region of fluid-film lubrication, the effect of various secondary factors, other than speed, load and viscosity as already discussed, on the position of the characteristic curve of $f$ versus $z\,N/p$ is as follows:

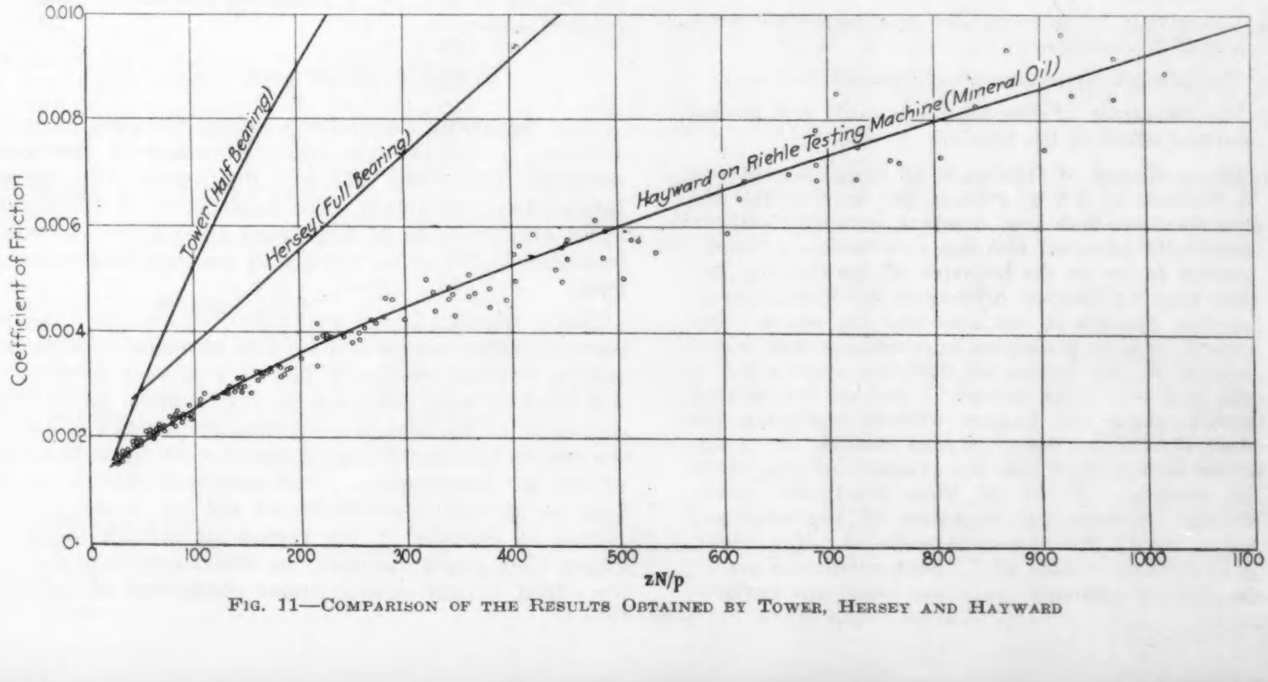


Fig. 11—Comparison of the Results Obtained by Tower, Hersey and Hayward

(1) It is unaffected by the method of oiling, provided enough oil is supplied to keep the bearing full

(2) It is unaffected by any variation in the "oiliness" of the lubricant, except possibly when the point of film rupture is approached very closely

(3) The curve is higher at small clearance-ratios than at large ones, but the differences are by no means so great as predicted by the general theory of lubrication for *ideal* bearings

(4) End-leakage tends to give abnormally high friction for bearings that are very short or have very large clearance

(5) The presence of oil-grooves in the pressure side of the bearing tends to give curves that are markedly convex upward, with abnormally high values in the intermediate range of $zN/p$. This is due unquestionably to interference with the normal building up of the fluid film in this range. In the absence of such grooves the lines are generally nearly straight, although they do not pass through the origin

(6) Within the limits of good practice, the character of the bearing metal does not affect the location of the characteristic curve

IV—The position of the critical point is very important, since this largely determines the safe operating value of $zN/p$, and hence the working value of $f$. Apparently, it is affected as follows:

(1) Within the limits of good practice, the use of smaller clearances tends to lower the critical value more than enough to counterbalance the increase in the height of the curve in the fluid-film region

(2) The use of lubricants of high oiliness tends to lower the critical point, probably because the presence of an adsorbed semi-solid film on the metal surfaces helps to prevent rupture of the fluid lubricating film, even after it has become extremely thin

(3) Various bearing metals differ appreciably in their ability to maintain the fluid film at extremely low values of $zN/p$

The greatest chance for improvement in the design and lubrication of bearings appears to lie in a thorough study of the effect of these variables on the critical point of film rupture.

V—In the region of partial lubrication,

(1) Viscosity is by no means the most important property of the lubricant

(2) The oiliness factor comes very largely into play

(3) The character of the bearing metals has an important effect on the friction

(4) The coefficient of friction is no longer *necessarily* a function of $zN/p$, even if the bearing and the lubricant are both kept constant, because a delicate semi-solid adsorbed film has now become a fundamental factor in the behavior of the bearing, and this may be affected *differently* by corresponding inverse changes in the load and the speed. The case is exactly analogous to conditions that might prevail in the region of fluid-film lubrication if the load were high enough to deform the bearing metal, which can happen without rupturing the fluid film if the speed is high enough, or if the speed were so high that the evolution of heat fused its surface. Either of these conditions would change the essential character of the bearings; hence, results for the same value of $zN/p$ might give different values of $f$. Such conditions are so far outside ordinary operating conditions in fluid-film lubrication that they can be neglected in a practical treatment of the subject, but in the region of *partial* lubrication we are dealing with a much more delicate bearing surface in the form of this adsorbed semi-solid film, which is much more susceptible to high pressures and temperatures than is any bearing metal

By the same reasoning, the critical value of $zN/p$ may be somewhat different for different combinations of the load and the speed and may change with the temperature. There is undoubtedly a fairly wide region of load and temperature values within which the critical point and $f$ in the region of partial lubrication will be a definite function of $zN/p$ for a given bearing and lubricant, and experimental work should be undertaken to define these limits

VI—The simple equations for ideal bearings approximate more closely to the observed curves for actual bearings, especially at the larger clearances, than do the much more complicated equations derived by Sommerfeld on the basis of correcting for the eccentricity of the journal, but neglecting the important effects of end-leakage and surface roughness, which are calculable only with difficulty. Furthermore, the points of minimum friction calculated by Sommerfeld, on the basis of hydrodynamics only, bear no relation to the actual critical points of film rupture.

VII—The true carrying power of a bearing can be calculated readily, once the critical point is known. For example, if the critical point is found to come at $zN/p = 10$, the maximum pressure that the bearing will handle without rupture of the film is $zN/10$. In other words it is directly proportional to the speed and to the working viscosity of the lubricant, divided by the critical value of $zN/p$. The only *absolute* limit to the carrying power of a given bearing is, therefore, the pressure that will deform the bearing or seriously deflect the shaft.

VIII—The efficiency of a bearing should not, however, be judged on the basis of its carrying power nor by the minimum coefficient of friction at the critical point, but rather by the coefficient of friction obtained when it is operating with a reasonable factor of safety. The proper factor of safety will vary with conditions but as a general proposition *the efficiency of a bearing can be measured in terms of the value of f obtained when operating at the value of z N/p which is five times that at the critical point.*

### APPLICATION OF THE CONCLUSIONS

The foregoing conclusions bring out very clearly the futility, if not the absurdity, of many of the common methods of testing oils and diagnosing the causes of lubrication difficulties. The application of these conclusions and methods of treatment to practical lubrication problems is therefore worthy of further brief consideration.

Since practically all ordinary "oil-testing" machines operate in the region of fluid-film lubrication, they are in reality nothing more nor less than a very poor type of viscosimeter and can give no information as to the behavior of an oil that is not given much more quickly and accurately by the intelligent use of a Saybolt viscosimeter or similar instrument. This naturally raises the question as to why such machines are not customarily designed to operate in the region of partial lubrication, where they *could* measure the effect of the oiliness factor. The reason is that under conditions of partial lu-

brication there is inevitably abrasion of the bearing surfaces; this abrasion changes both the clearance and the smoothness of the surfaces and makes it almost impossible to get concordant results. The majority of bearing machines are therefore designed for and operated under conditions of perfect fluid-film lubrication to get reasonably reproducible results, apparently regardless of their lack of real significance.

Several investigators have realized the need for reaching or approaching this point of rupture, even at the expense of having to change or renew the bearings frequently. But almost invariably this point of rupture of the film has been reached by greatly increasing the loads. This necessitates heavy cumbersome apparatus, tends to deflect the journals or cause the bearing metals to flow and makes accurate measurements almost impossible. It would be much better to keep the load at a value approximating service conditions and to *reach the critical value of $z\ N/p$ by lowering the speed instead of increasing the pressure*. The film rupture can then be measured on lighter apparatus, with greater accuracy and with much less abrasion of the surfaces per unit of time.

Similarly, in determining the shape of the characteristic curve at very high values of $z\ N/p$, it is desirable in general to obtain these high values by lowering the load rather than by increasing the speed, since working at high speeds with the consequent heating effects[13] makes it difficult to measure accurately the working temperature and the viscosity of the lubricating film.

The present methods of diagnosing and attempting to cure lubrication troubles are extremely haphazard. The customary rule of using the oil of the lowest viscosity that will operate satisfactorily has, of course, a fundamentally sound basis, but there is no definite criterion for choosing oil except by the method of trial and error. The fact is often overlooked that *every* method of lowering the coefficient of friction, either by reducing the speed, the viscosity of the lubricant or the area of the bearing, accomplishes its effect by bringing the bearing closer to conditions where $z\ N/p$ has a very low value and where the film is likely to be ruptured and the bearing injured. In other words, as is frequently the case in engineering work, these savings are accomplished by the simple expedient of reducing the factor of safety. If the factor of safety is unnecessarily high, this is desirable, but the essential point to determine is *what factor of safety is necessary* for given conditions of operation. It is then a comparatively simple matter to choose the lubricant to give neither more nor less than this proper factor of safety. For a given bearing, working under given conditions, there is a *safe* value of $z\ N/p$ that will give a reasonably low coefficient of friction and the viscosity of the lubricant should be selected to give this proper value.[14]

Where the bearings are those of line-shafts, countershafts and machine tools that are started and stopped rather frequently, subjected to peak loads, not given constant attention to prevent the temporary failure of the

[13] It should be noted that equal values of $f$ corresponding to equal values of $z\ N/p$ do not necessarily mean equal amounts of heat evolved. The power lost in a given bearing equals the product of the factors $k$, $f$, $p$ and $N$. If we assume as an approximation that $f = K_1\ (z\ N/p)$, the lost power will equal $k_2\ (z\ N^2)$, which is far from being a function of $z\ N/p$. This does not in any way invalidate the general proposition that the operating value of $f$ is the best single measure of the efficiency of a given bearing for a given purpose, because the speed of the journal in revolutions per minute is almost invariably fixed by other considerations.

[14] This calculation is, of course, complicated by the necessity of knowing the approximate operating temperature of the bearing for the conditions in question. This generally can be estimated with sufficient accuracy except at high speeds. Even then, if a part of the energy now devoted to determining the high-speed coefficients of friction for a given lubricant in a given bearing, the data for which can apply only to the precise combination on which it was determined, were directed toward the much simpler problem of measuring and deriving formulas for the *rate of dissipation of heat* as a function of the temperature of the lubricating film, the calculation of the equilibrium operating temperature would be merely a matter of finding at what temperature the rate of heat generation in British thermal units per hour, which equals $0.202\ f\ p\ N\ D^2\ L$ (where $f$ is calculated from the corresponding value of $z\ N/p$, and $D$ and $L$ are given in inches) just balances the rate of heat dissipation. The rate of heat dissipation equals $k\ (T_b\text{-}T_r)$, where $T_b\text{-}T_r$ is the difference between the temperature of the bearing and that of the room and $k$ is the coefficient of heat transfer for the bearing in question. Stodola has determined such data for steam-turbine bearings.

TABLE 1—VALUES OF $z\ N/p$ FOR VARIOUS RECOMMENDED CONDITIONS OF OPERATION

| Location of Bearing | Lubricant | $p$ | $N$ | $z$ | $z\ N/p$ | $c/D$ |
|---|---|---|---|---|---|---|
| Automobile Crankshaft | Medium Machine Oil | 300 to 700 | 900 to 1,400 | 7 to 8 | 15 to 25 | <0.0010 |
| Aeronautic Engine Crankshaft | Heavy Engine Oil | 300 to 1,800 | 1,800 to 2,000 | 7 to 8 | 15 to 25 | <0.0010 |
| Stationary Gas Engine Main | Medium Machine Oil | 500 to 700 | 250 to 800 | 30 | 25 | 0.0010 |
| Stationary Gas-Engine Crank-pin | Medium Machine Oil | 1,500 to 1,800 | 250 to 800 | 50 | 15 | <0.0010 |
| Stationary Gas-Engine Cross-head | Medium Machine Oil | 1,500 to 2,000 | 250 to 800 | 40 | 10 | <0.0010 |
| Diesel Engine Main | Heavy Engine Oil | 250 to 600 | 60 to 160 | 30 | 15 | 0.0010 |
| Diesel Engine Crankpins | Heavy Engine Oil | 1,500 to 4,000 | 60 to 160 | 40 | 2 to 5 | 0.0010 |
| Marine Steam Engine Main | Machine Oil | 275 to 500 | 180 | 30 to 40 | 20 to 30 | <0.0010 |
| Marine Main Crankpin | Machine Oil | 400 to 500 | 180 | 30 to 40 | 20 | 0.0010 |
| Stationary Slow-Speed Main | Heavy Machine Oil | 80 to 400 | 40 to 80 | 70 | 20 | <0.0010 |
| Stationary Slow-Speed Crank-pin | Heavy Machine Oil | 800 to 1,300 | 40 to 80 | 80 | 6 to 8 | <0.0010 |
| Stationary Slow-Speed Cross-head | Heavy Machine Oil | 1,000 to 1,500 | 40 to 80 | 70 | 5 | <0.0010 |
| Stationary High-Speed Main | Engine Oil | 60 to 250 | 360 | 15 | 25 | <0.0010 |
| Stationary High-Speed Crank-pin | Machine Oil | 400 to 1,500 | 360 | 30 | 6 to 15 | <0.0010 |
| Stationary High-Speed Cross-head | Machine Oil | 1,500 to 1,800 | 360 | 25 | 5 | <0.0010 |
| Locomotive Drive-Wheel | Heavy Machine Oil | 550 | 250 | 100 | 30 to 50 | <0.0010 |
| Locomotive Crankpin | Heavy Machine Oil | 1,500 to 2,000 | 250 | 100 | 5 to 8 | <0.0010 |
| Locomotive Cross-Head | Heavy Machine Oil | 3,000 to 4,000 | 250 | 130 | 6 to 8 | 0.0010 |
| Marine Steam Turbine | Light Machine Oil | 85 | 2,000 | 10 | 250 | 0.0010 |
| Stationary Steam Turbine | Machine Oil | 400 to 950 | 2,000 | 20 | 100 to 200 | 0.0010 |
| De Laval 7-Hp. Steam Turbine | Light Machine Oil | 7 to 15 | 30,000 | 1 | 1,500 to 3,000 | 0.0020 |
| De Laval 300-Hp. Steam Turbine | Light Machine Oil | 20 to 25 | 10,500 | 2 | 1,000 | 0.0020 |
| Railway Car Axle | Heavy Machine Oil | 300 to 450 | 300 | 100 | 50 to 100 | |
| Generator and Motor | Engine Oil | 30 to 80 | 150 to 500 | 25 | 200 | 0.0010 |
| Rolling Mill Main | Hot Neck Grease | 1,800 to 2,500 | 60 | — | <1 | — |
| Cotton Mill Spindle | Spindle Oil | 1 | 8,000 to 12,000 | 2 | 10,000 | 0.0050 |
| Gyroscope | | 750 to 850 | 800 to 1,500 | 60 to 30 | 55 | 0.0013 |

oil supply, and the like, a fairly high factor of safety, probably around 15, is required. For continuously loaded bearings that are better cared for, such as generator and turbine bearings, a factor of safety of 5 should be adequate.

Where the direction of the application of the load is changed during every revolution, it is much easier to prevent the rupture of the lubricating film as there is insufficient time for the oil to be squeezed out from the region of high pressure. Small clearances are of great assistance in this respect and, under these special conditions, well designed bearings may operate satisfactorily and give very low coefficients of friction by working at values of $z\ N/p$ that are very close to the critical point.

The use of these factors of safety obviously requires more definite data as to the location of the critical point for different types of bearing. Some indirect evidence on this point can be obtained by calculating the operating values of $z\ N/p$ corresponding to the best practice. The values in Table 1 are calculated primarily from the recommendations in Alford's book on Bearings and Their Lubrication.[15]

Another important practical application of the $z\ N/p$ concept is in connection with diagnosing the fundamental cause of lubrication troubles. The method is especially helpful in studying the behavior of the special types of bearing frequently but incorrectly termed "oilless," such as genelite, and the like.

Suppose a particular bearing with a certain oil is giving trouble, presumably due to an abnormally high coefficient of friction. As has been indicated, such high coefficients may be caused by a variety of conditions, the cure for each of which is different. If, however, $z\ N/p$ is calculated for the condition in question, it will be found that the case falls within one of the three classes that follow:

(1) In the case where $z\ N/p$ is very low, say below 10, the difficulty almost certainly is due to operating under conditions of partial lubrication, and the simplest cure is to increase the viscosity of the lubricant to give a higher value of $z\ N/p$. Some improvement can be obtained also by using a lubricant having more oiliness, or possibly by changing the metal in the bearing to make the point of rupture somewhat lower. In general it is unsafe to operate with such low values of $z\ N/p$

(2) In the case where $z\ N/p$ may be very high, say above 600, the difficulty can be overcome most readily by decreasing the viscosity of the lubricant, or by decreasing the area of the bearing and hence increasing the pressure thereon

(3) In the case where $z\ N/p$ is found to lie in a proper intermediate range, say between 25 and 200, the abnormally high coefficient of friction can be due only to highly abnormal conditions in the bearing, such as a poorly aligned shaft, too tight bearings, or improper methods of feeding the oil that do not keep the bearing full. These possibilities must be investigated and this generally can be done readily without complicating the case by trying out other lubricants of different viscosities that could only aggravate the situation

The value of ball and roller types of bearing lies, not in their giving coefficients as low as those obtainable near the critical points for good journal bearings but in the fact that they give a very flat curve of moderate height, with no critical point and sharp rise in the coefficient at low values of $z\ N/p$.

[15] See bibliography at end of paper.

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# Progress Made in Garage Equipment

*By* H. C. Buffington[1]

Chicago Service Meeting Paper *Illustrated with* Photographs

THE paper relates specifically to the type of garage equipment that is used to handle the motor vehicle in preparation for its repair. The devices illustrated and described are those designed to bring in disabled cars, and include wrecking cranes and supplementary axle trucks; portable cranes and jacks on casters for handling cars in a garage; presses, tire-changing equipment and wheel alignment devices; engine and axle stands; and miscellaneous minor apparatus.

The different factors mentioned emphasize the great need of standardization. The thought is not to do away with a car's individuality, but to construct all parts so that cars may have efficient service to the highest degree through the agency of every serviceman.

IT is interesting to note that during 1921 alone over 10,000,000 cars and trucks were registered in the United States. It is safe to say that during 1922 another 1,000,000 cars will be added. Does it seem any wonder then that the service requirements throughout the entire automotive field are demanding such careful attention? Actual practice, resulting from the scientific study of the automotive service problem has at last reached a point where it is possible to handle automobiles in a satisfactory manner. This fact is the result of the gradual development of a general standard design. However, there is yet much standardizing to be done in details that will help to simplify the service work. Therefore, it is from the manufacturing standpoint of service equipment that I shall make the following suggestions, briefly touching upon the progress already made in garage equipment. There are, speaking approximately, two kinds of garage equipment; (*a*) that which actually makes and repairs the necessary parts and (*b*) that which handles the automobile in preparation for repair. It is this latter class that I wish to discuss.

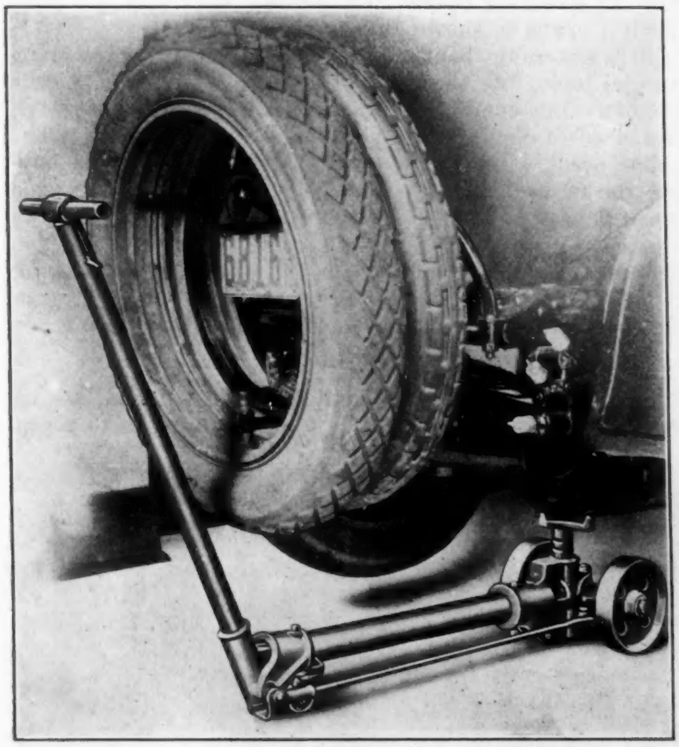

Fig. 1—Jack for Lifting One End of a Car Clear of the Floor and Then Moving It to Enable the Mechanic To Get at Any Part Needing Attention

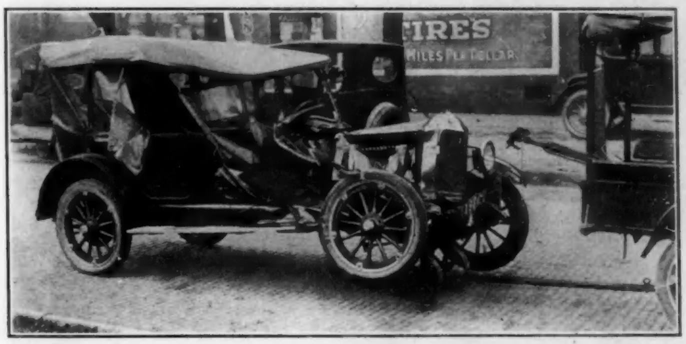


Fig. 2—Device for Bringing In Disabled Cars

One of the first problems that confronts the repairman is lack of floor space. When we consider that a car covers approximately 90 sq. ft. of the floor, we realize the necessity of contriving to overcome this. It was with this in mind that the jack shown in Fig. 1 was developed. This can be placed under the front and rear ends and, by caster wheels, the car can be pushed in any direction, thus enabling the mechanic to get at any part of it; he also can transport it to any part of the garage or the yard when desired. The next problem that arose was

Fig. 3—Wrecking Crane Lifting a Car Out of the Ditch

[1] M.S.A.E.—Chief engineer, Weaver Mfg. Co., Springfield, Ill.

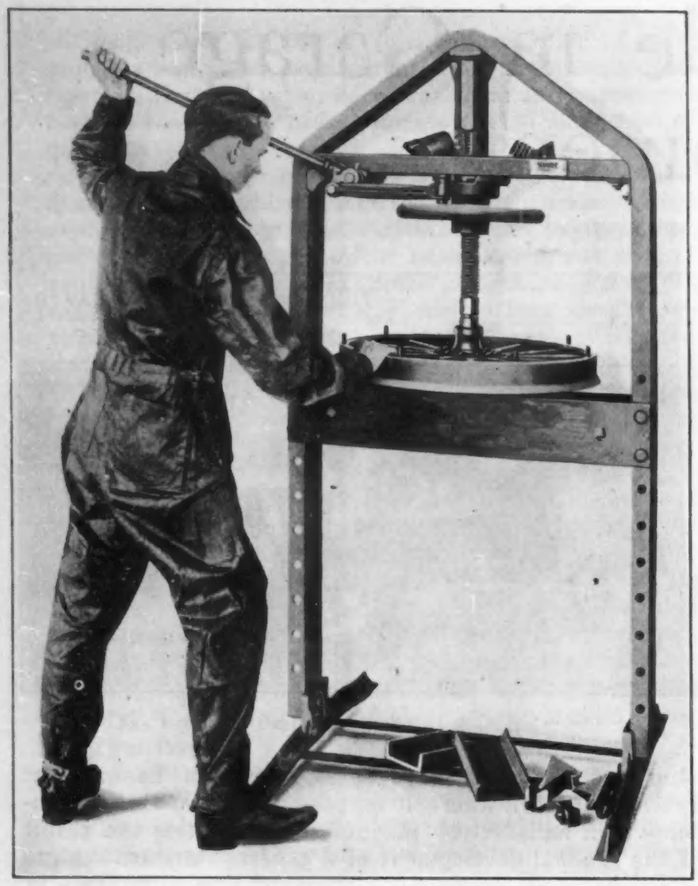

FIG. 4—PORTABLE PRESS OF THE SCREW TYPE FOR STRAIGHTENING BENT PARTS

FIG. 5—TIRE CHANGER THAT IS EQUIPPED WITH ATTACHMENTS FOR HANDLING PRACTICALLY ALL SIZES AND TYPES OF TIRE

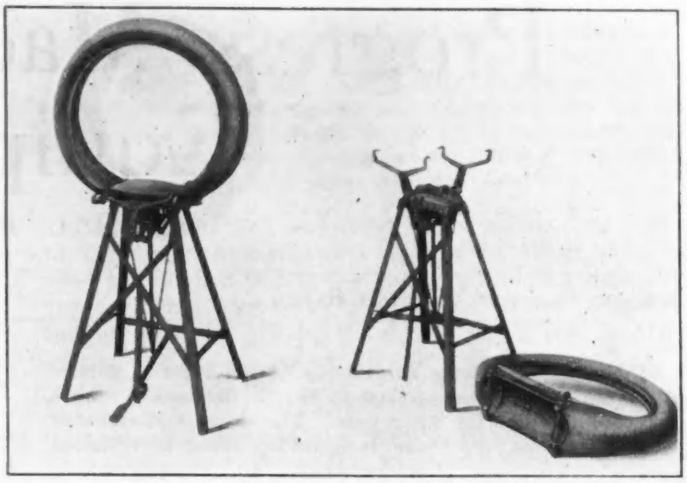

FIG. 6—TIRE SPREADER THAT NOT ONLY SPREADS THE CASING TO ENABLE REPAIRS TO BE MADE, BUT ALSO TURNS IT INSIDE OUT AT ANY DESIRED POINT FOR INSPECTION

the lack of a contrivance to bring in a disabled car, especially if a wheel or an axle were broken. After considerable experimenting, a two-wheeled device such as is shown in Fig. 2 was developed, having a long telescoping pole. It proved successful and thousands of wrecked cars have been brought into the garages by this means. Much development work has been carried on with the wrecking crane, and both very simple and very elaborate types have been brought out. This resulted from the difficulty experienced in rescuing ditched or overturned cars. Many garages are making a special feature of operating wrecking cranes to increase their repair business. Such a crane is shown in Fig. 3.

Straightening bent parts is another task requiring special tools. As the work of putting wrecked cars back into running condition increased, the garage owner felt the need of having the work done in his garage. The arbor press was not powerful enough and a press having the necessary power was too heavy and occupied too much floor space. Therefore, a portable press having the necessary power was designed, as illustrated in Fig. 4. This press is of the screw type and is adjustable, making it possible to handle either large or small work. It can be used in either a vertical or a horizontal position.

### TIRES AND WHEELS

Few garagemen pay enough attention to the loss of time in changing tires. It is a common sight to see a

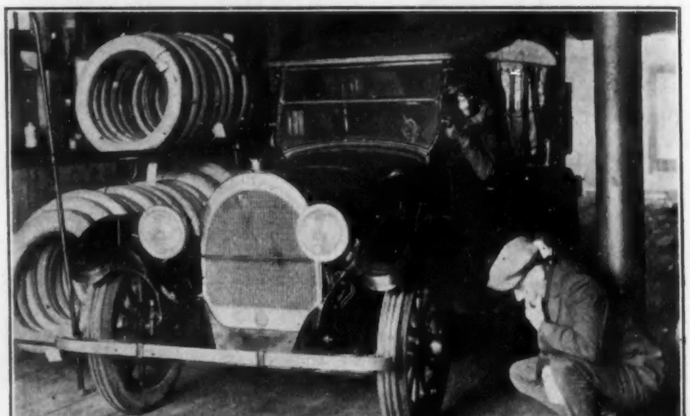

FIG. 7—WHEEL ALIGNMENT INDICATOR

rim battered and bent out of shape. A successful tire-changer, as shown in Fig. 5, has now been put on the market. It is supported on a stand of suitable height, and is provided with different attachments for putting on or removing the different types and sizes of tire. In touching upon the subject of tires, the development of tools and fixtures for the handling of tires brings home very forcefully the desirability of paying closer attention to the adoption of standards, especially where it is evident that replacements are necessary. The automotive engineer should have in mind two things if he expects his automobile to be popular; (*a*) he must adopt existing standards and adhere to prevailing designs

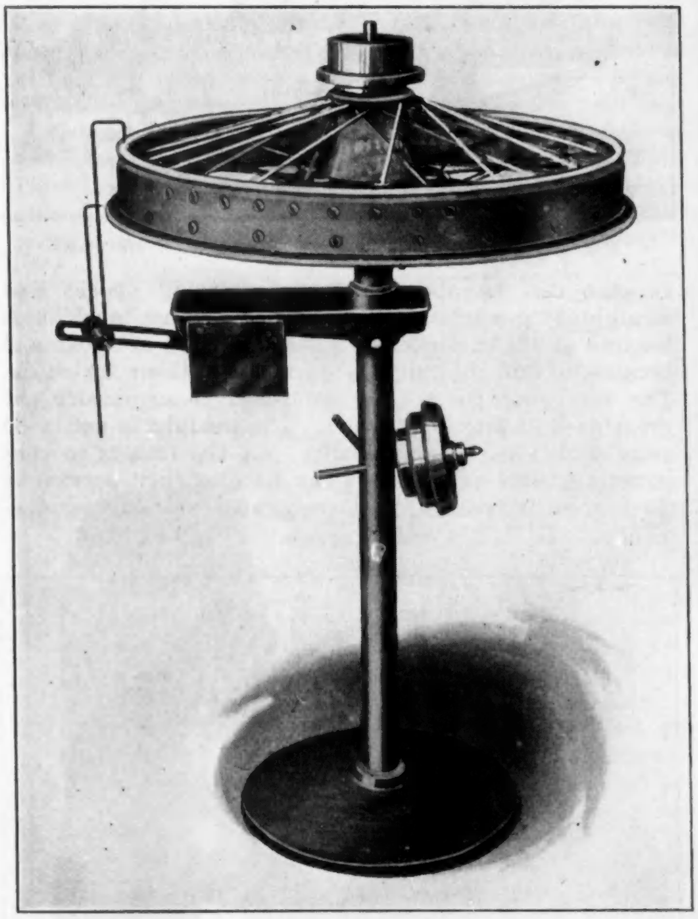

Fig. 8—Wire-Wheel Stand

wherever there is wear and (*b*) he must build his individuality around the features that, for ordinary use, need no replacement. For example, we have the clincher, the quick-detachable and the split-rim types of tire attachment. All of these are effective in holding the tire in place. Individually, any one of them would do the work, even if the others were not in existence. The garageman is familiar enough with any one of the types, but his difficulty lies in his need of an inexpensive tool that will remove and replace the tire without requiring the adjustment of attachments to meet the needs of the different types of tire. Standardization of tire types will be a progressive step in making for better service.

After the tire-removing machine was brought out, it was seen that it often was advisable to inspect the inside of the casing when a tire had been removed. However, the casing is very stiff and it requires some effort to examine it properly. Fig. 6 shows a tire spreader that, in addition to spreading the casing for repairing the

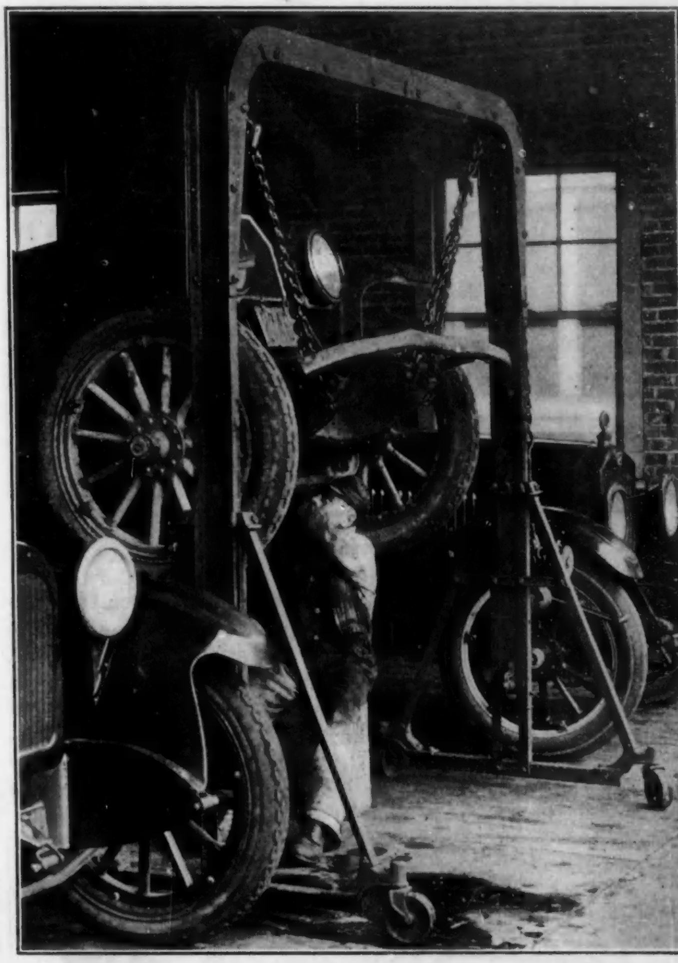

Fig. 9—Portable Hoist That Will Do Practically Everything That a Traveling Crane Will and in Addition Can Be Pushed Aside with the Car Suspended in the Air if Necessary

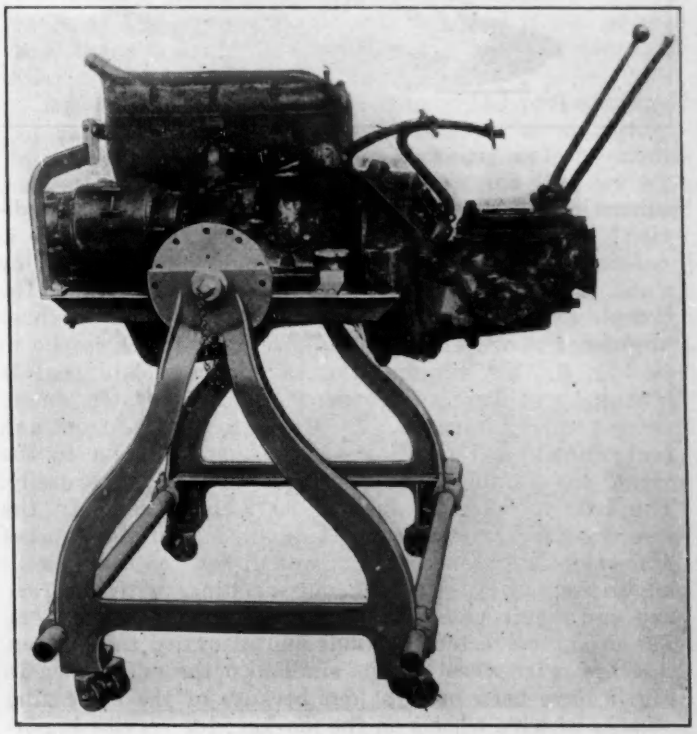

Fig. 10—Stand That Facilitates Engine Repairing

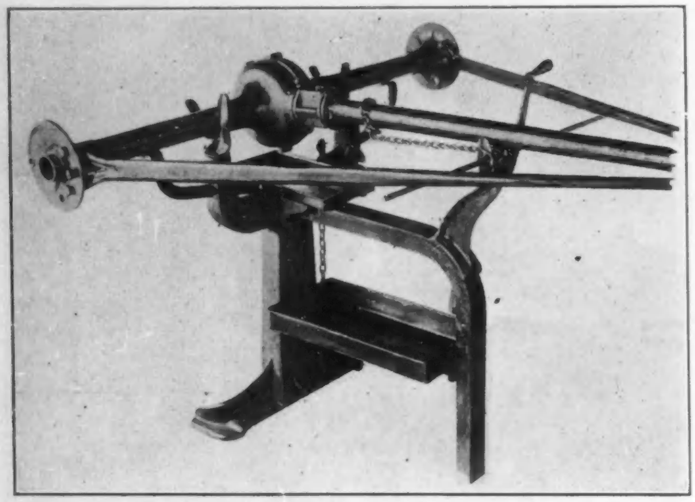

FIG. 11—STAND FOR HOLDING AN AXLE WHILE REPAIRS ARE BEING MADE

tire, has the advantage of permitting one to see the condition of the casing. Besides spreading the casing, the machine serves to turn it inside out at any desired point. Hundreds of rim removers and tire spreaders have been put on the market, and this points out the need of this equipment to the repair-man.

How many car-owners, or salesmen even, know why one tire will often wear out much more quickly than the

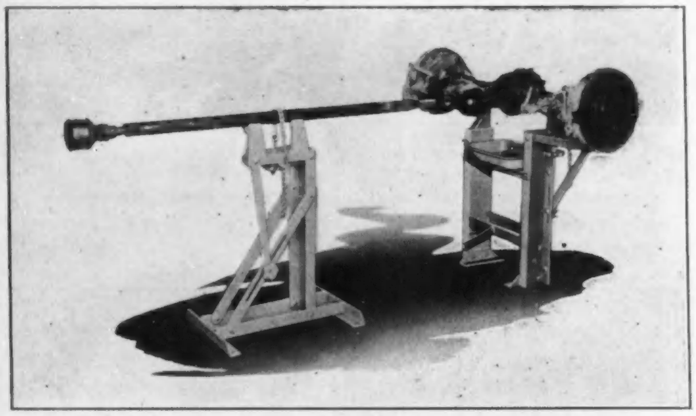

FIG. 12—ANOTHER FORM OF AXLE STAND

other three? How many know how to correct this condition? The first time one will doubtless think the tire is defective. How much better it would have been if, by some simple method, one could have discovered the trouble and remedied it. An indicator to show the misalignment of wheels has been developed, which is shown in Fig. 7. By running the car over a plate that is mounted on rollers and connected to an indicator, an incorrect setting is gaged. By straightening the front and rear wheels, and sighting across from the rear to the front, the trouble sometimes can be seen very easily. The trouble often is due to the unusual wear in the steering-knuckles and connecting links. In the process of developing the indicator, it was discovered that a well-known factory was sending out machines with the rear axle out of line, causing unusual wear on one of the tires. The dealer corrected the fault and informed the builder.

A few wire-wheel stands similar to the one shown in Fig. 8 have been brought out because of the increasing number of wire wheels on the market; the service importance of such a stand has not been appreciated fully. The wire wheel is another example of extravagant waste of service energy and, because of lack of standardization, it is suffering unpopularity, not alone from the owner's standpoint but from that of the service-man as well. This is due to the spoke, a simple little wire upset on one end and having a thread on the other. Before the wire wheel will enjoy the popularity it deserves, every wheel maker must adopt a standard spoke, with the same "crooks" and the same lengths for different sizes of wheel. A good

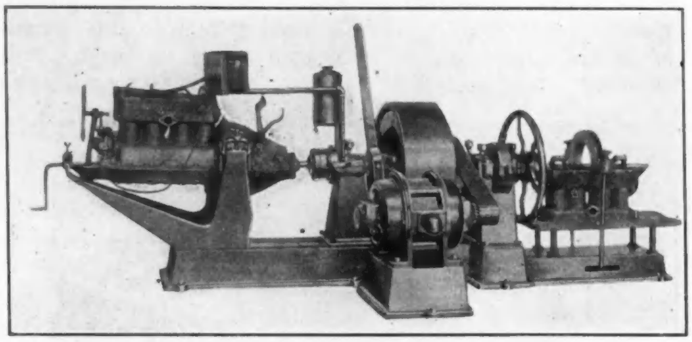

FIG. 13—STAND FOR RUNNING-IN NEW ENGINE BEARINGS

revenue can be obtained from replacing spokes and straightening wheels, but repair-men will not touch them because of the confusion of sizes and types of spoke and because of his difficulty in purchasing them for stock. The foregoing points are mentioned to emphasize the great need of standardization. The thought is not to do away with a car's individuality, but the idea is to construct all parts so that cars can have efficient service to the highest degree, through the agency of every service-man.

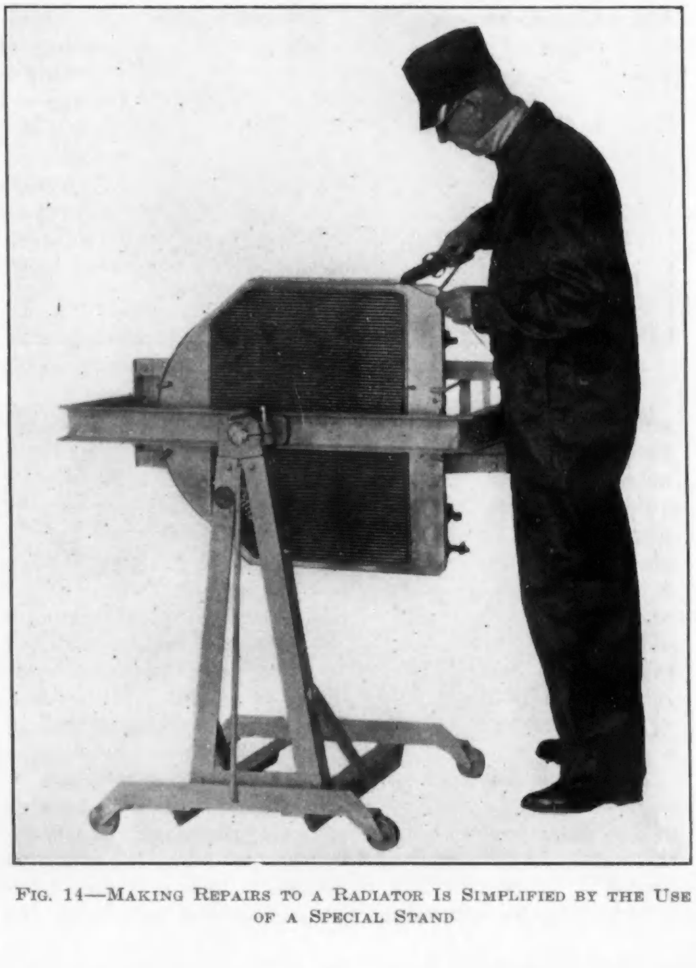

FIG. 14—MAKING REPAIRS TO A RADIATOR IS SIMPLIFIED BY THE USE OF A SPECIAL STAND

Perhaps one of the most useful pieces of garage equipment is the portable hoist shown in Fig. 9. It performs every operation that a traveling crane can perform except the lifting of the whole car. It can be pushed aside with the car partly suspended in the air if the repair work is delayed.

### Hoists and Jacks

In mentioning the portable jack, our attention has been called to the fact that a car should not be jacked up under the differential housing, because some axles are built of pressed steel or aluminum. Engineers who think that a car is not jacked up at this point will do well to realize the desirability of placing a special flat pad there; it is from this place that a car is always lifted. Another point is that of ground clearance of the front and rear axles. It seems that a standard minimum height should be established that will permit the design of a jack having a reasonable amount of lift. Our attention was called recently to a taxicab that had only a 7½-in. clearance from the ground and called for a special jack. The portable jack is used with hardly an exception in every garage. It is necessary for quick service. When there is practically no room underneath for the jack, the delay in lifting the car is apparent. A jack built for the height previously mentioned, would not be practical for general use.

Fig. 15—Aligning a Piston and Piston-Rod

### Engine and Axle Stands

The automobile engine has remained a mystery to the average man for so long a time that the designer still thinks he is privileged to keep away from conventional design. The engine designer does not appreciate the convenience of handling an engine in an engine stand. There are different numbers and sizes of cylinders, but it is important that a standard fastening be adopted. It would be well if an extra pad or arm could be installed at some convenient place on the crankcase. It would not cost any more, as it is done in the manufacture of engines to accommodate the tools and fixtures. There are a number of cleverly constructed engine-stands on the market, an example being shown in Fig. 10, which have increased the efficiency of overhauling and repairing engines wonderfully.

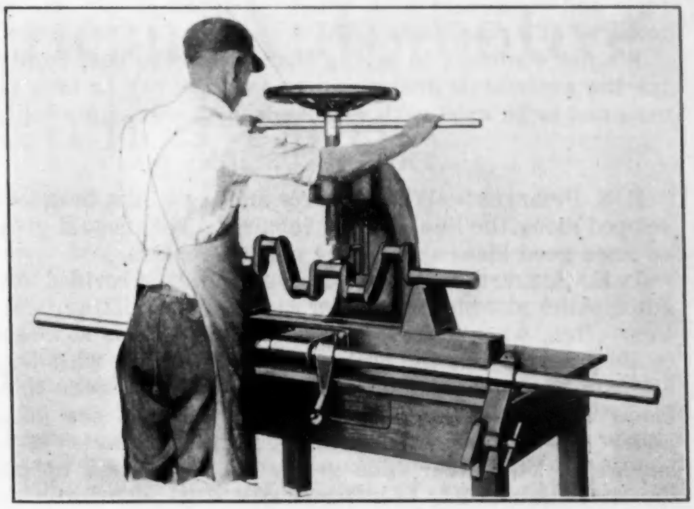

Fig. 16—Bench Press

Another useful device that has come into general use is the axle stand, shown in Figs. 11 and 12. It holds the axles rigid while the work is being done on them. Garage equipment for many other purposes has been brought out, such as the running-in stand for running-in new engine bearings, shown in Fig. 13; the radiator stand, Fig. 14; the truck-wheel dolly for moving truck wheels; and various small tools and devices, some of which are shown in Figs. 15 and 16.

The application of oils and greases has received much attention from the service-men. Aside from the various makes of grease guns and machines, one of the most familiar pieces of equipment about the garage is the oil-bucket pump shown in Fig. 17. This pump is arranged so that it will draw out the old oil as well as supply new lubricant and, at the same time, measure the amount put in. As a rule, engineers have always realized the importance of accessibility in oiling a car, and it can be seen that there is an improvement in the oiling system each year.

Although I have only touched upon the progress made in garage equipment, I believe this to be an entirely new angle in approaching the subject. I wish to emphasize the necessity for standardization and that we wish to make it possible for the repair-man to obtain standard

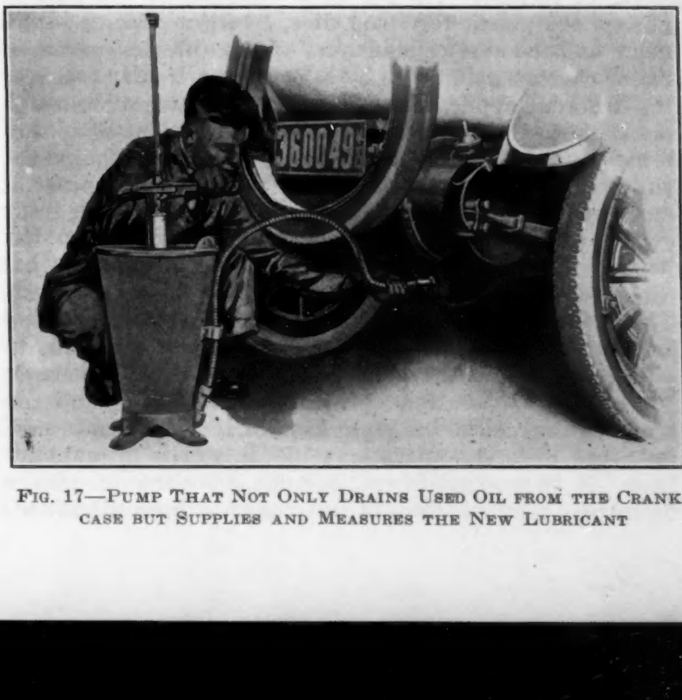


Fig. 17—Pump That Not Only Drains Used Oil from the Crankcase but Supplies and Measures the New Lubricant

tools and equipment with which all types of car can be repaired at a reasonable price.

We feel confident in saying that one of the best points for the automobile dealer will be that the car he sells is designed to be used with all standard garage equipment.

## THE DISCUSSION

B. S. PFEIFFER:—What fixtures and tools have been developed along the line of Ford repairs? They would give us some good ideas about more general repairs.

J. E. SCHIPPER:—Some place should be provided on automobiles at which the owner can apply the lifting jack. Very often, when a front tire is flat, the axle is so close to the ground that the jack which is provided with the car cannot be placed under the axle. When a rear tire blows out, the average person will try to put the jack under the spring; the spring generally deflects just enough to let the car slide gently off of the jack about the time the rim has been taken off. Provision for lifting-jack application is one feature that has been overlooked. There ought to be some kind of a projection, so that a jack can be placed under it and the car raised high enough to facilitate changing the tire.

A. H. PACKER:—The question of maintaining the car or truck in operative condition depends upon the electrical as well as the mechanical features because, according to the records of the Chicago Motor Club, a greater percentage of the troubles which tie up a car are of an electrical nature. There is a need for good, low-priced electrical-testing equipment, and a need for education. Regardless of how the men can get the education, service-stations will have a difficult time in handling the service work with men who do not understand their work, particularly in regard to the electrical equipment. There are several electrical schools that are making it possible for electrical work to be performed intelligently. Therefore, the equipment lines should be extended to include magnetizers, testing equipment, test benches and armature testers. A number of high-grade manufacturing companies are putting out that type of equipment. Another type of equipment makes a spectacular test that means little, if anything. A man who ties up money in equipment of that kind will find it comparatively useless and, eventually, this will reflect against the makers of legitimate apparatus and increase the resistance that their salesmen meet. In developing or exploiting any device or apparatus for electrical or mechanical testing, I think it should be carefully analyzed to make sure that it is legitimate and intended to facilitate service.

W. C. ALLEN:—Having been engaged in the sale of garage equipment for some time, I believe that the engineers and the service managers of the different makes of motor car can gain much by selecting particular tools for the different applications that are made and recommending them to their dealers. The present figures show that somewhere near $68,000,000 worth of garage equipment was sold during 1921. I believe that 40 per cent of it has been discarded since. The car dealer or garage owner, in trying to create more efficient methods has bought different sorts of tools and, after finding that certain tools did not do the work, he discarded them. The previous remarks concerning good electrical-testing devices that are made to use and others that are made to sell apply also to the garage-equipment field. There is not enough cooperation between the car dealer and the builder in regard to the right kind of tools needed to perform the different operations in the repair of automobiles. If 40 per cent of this $68,000,000 worth of garage equipment that was bought last year has been discarded, somebody must pay for it, and this usually reverts to the consumer. There is much room for improvement in the selection of proper tools to give the customer the most economical service.

MR. PFEIFFER:—It is the lack of standardization between other makes of car that gives the Ford service-station the advantage over the ordinary garage-man. When a garage does repair work on 100 different makes of car, no two of which are alike in even major dimensions, it cannot install special tools and fixtures for each one. The Ford service-station can and does this very thing and, after a number of years, the company has developed a system of standardized flat prices for a job. The total investment in this specialized equipment is not large, and the company accomplishes wonders with its help. Its service organization is trained as it must well be. I recall a Ford dealer who used one man for 8 hr. to overhaul an engine. This consisted of taking the engine out of the car, tearing it down, reboring the cylinders, fitting new pistons and rings, grinding the valves and putting in new main and connecting-rod bearings; in other words, it was a complete job. How long would it take an ordinary garage to do the same job, even on a Ford car? Most of this equipment has been designed and developed by the Ford engineers. It seems that other companies should give this important matter their attention. Anything that enables the dealer to give better, quicker and less costly service will help him and, in turn, the manufacturer.

G. A. TOAZ:—Lifting jacks are selected usually by the engineers, and I think the responsibility should rest there. I know of one very recent instance in which a car stripped a tire. The driver could not replace the tire himself because the cogs on his lifting jack would not hang together. The interest the engineer has in the service end is not active enough. We must remember that we are not only selling cars but selling service. If we do not make proper provision for that service in the tools actually sent with the car, we are not doing what we should do. In selecting the tools to be supplied with the car, we should provide usable tools or omit them from the list entirely.

H. C. BUFFINGTON:—The company I represent builds this equipment because there is an actual demand for it. It has worked on many devices but it does not put them on the market until there is a demand for them. It was said previously that much of such equipment is not used, but there are many tools that go with a car that are not used. The closer we can approximate a general standard car, the easier it will be for every garage-man to handle the car. There will then be no need of a special service-station for every special car. The special service-station does not exist in small towns. In a small town, if a tool can be purchased that will do general work on all cars, it will sell.

B. M. IKERT:—Concerning the small-town garage, in a certain Western town of about 100,000 population, one service-station handles a very high-class car and did not get service orders on more than 20 per cent of the cars known to be in its territory. I could see the reason after taking one look at the shop and its service methods. It had very little equipment and no system whatsoever. Consequently, the owners were not taking their cars there. In another town of 2800 people, a man has a shop and the agency for one popular make of car. He can do everything in that shop but cut gears, as he has grinding, drilling and shaping machines. There are many tractors

(*Concluded on page* 80)

# European and American Automotive Brake and Clutch Practice

*By* H. G. Farwell[1]

Metropolitan Section Paper *Illustrated with* Photographs and Diagrams

THE author describes the major features of brake and clutch practice that he observed in 1920 while traveling in England, Belgium, Italy and France, comparing them briefly with American practice of the same period. He analyzes the types of brake and clutch used on 165 cars exhibited at the London automobile show of that year, giving the percentage of the different types in evidence.

Numerous illustrations that are described and commented upon in greater or less detail appear in the paper and in the discussion which followed it, these being inclusive of most of the best-known types of brake and clutch in use in the United States and in Europe.

BEFORE recounting the major features of European brake and clutch practice that I observed during several months of travel in England, Belgium, Italy and France in 1920, I will outline the general practice in the United States as it existed at that time. The ordinary type of external brake was in general use for service work; in Europe it is known as the foot-brake. What we call the emergency brake is known as the hand-brake in Europe because it is operated by the hand-lever; it is nearly always cam-operated in this Country. There were also some cases in which one brake, sometimes the service and sometimes the hand-brake, was used on the transmission; cases in which both sets of brakes were internal, on the rear wheel-drums; cases where these were in concentric drums; and, in some instances, the two brakes were placed side-by-side. Regarding clutch practice in the United States, we had seen a gradual reduction in the number of cone clutches used, until comparatively few cars are so equipped in proportion to the number of cars built. We have seen a gradual increase in the use of the disc clutch; the single-driven-plate and multiple-disc types being used in high-grade cars. One striking feature is the size of brake-bands. In this Country the diameter of the rear-wheel brakes usually is from 35 to 45 per cent of that of the rear wheels. It is not unusual to find 12, 14 and 18-in. drums on cars weighing from 2100 to 5500 lb.

## England

In England the use of the internal brake was fairly general on all classes of cars and, with comparatively few exceptions, the use of the transmission brake was rather more marked there than in this Country. In general, very little criticism was found on the operation of these brakes, but there is one exception in the case of the Daimler car which, until recently, has always used an external service-brake on the rear-wheel drums. I understand that the company is planning now to change that to the conventional internal expanding type. On several of the post-war cars weighing 1600 lb. and even less, they were using external band brakes. The objection made in England to the use of the external brakes is that they always drag. Their engines are very much smaller, for fuel-economy reasons; any dragging of brakes uses up a greater proportion of available engine power than is the case with cars in the United States.

The usual English criticism of American brakes is that they are intolerable, although I found about as many unsatisfactory brakes in England as here, on from one to five makes of car, depending upon the locality in which they are used. The brakes on city cars generally are maintained better than the average brake used in the country districts. Probably more attention has been given in England to brake design and the construction of internal brakes than here; in fact, that is true throughout Europe. The main objection to the usual internal brake is that wear is confined to a very much smaller area than is the case with external brakes. To compensate for this, the English designer arranges the adjustment feature so that brake adjustment can be made much more easily, often without getting out and going under the car. In the English models that have been produced within the past two years, the ease of brake adjustment has received great attention. In some cases large hand-wheels are placed so that one can adjust the angle of the cam with but little more difficulty than that of getting out of the car.

I had been led to believe that metal brakes were always noisy, but I found this to be an exaggeration. Except for the metallic grind, these internal brakes were only slightly more noisy generally than the American conventional type of external brake. I attribute this to better or closer fitting and to the elimination of points where vibration may occur. One finds noisy brakes in England, the same as here; but, in general, they are not much worse there. Brake-drums are very much smaller in England. To cite one case, an American car weighing approximately 2100 lb. was equipped with 31 x 4-in. tires and 14-in. brake-drums. The English car, having exactly the same size of engine, weighed 2800 lb., had 30 x 1½-in. tires and the diameter of the brake-drums was 10 in. After looking over the field, I am sure that the tendency in this direction is to increase the diameter of the brake-drums. Probably tire sizes will always be kept smaller, for the sake of economy and first cost, but the brake-drum diameters are certainly growing.

The English clutch situation was interesting in that it showed so many cone clutches on cars of the higher grades. The single-plate clutch also is used, but not to the same extent as here. There is a noticeable tendency to consider the use of the plate clutch, instead of the cone type. It will be surprising to see the great number of plate clutches that will be used in the years to come. There is less need for large-capacity clutches in England than there is here. That has a bearing on the use of the multiple-disc clutch. It is interesting to note some of the tendencies at the automobile show held in London in

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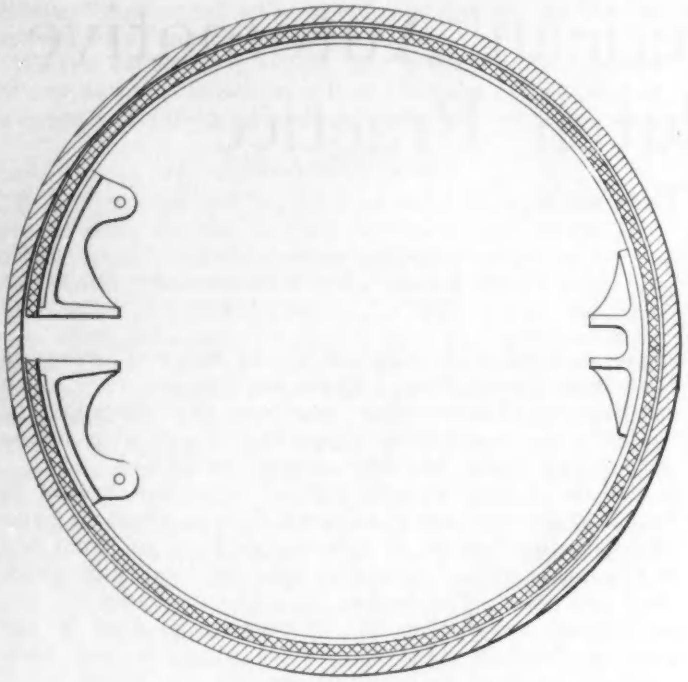

FIG. 1—CONTINUOUS-BAND TYPE OF BRAKE

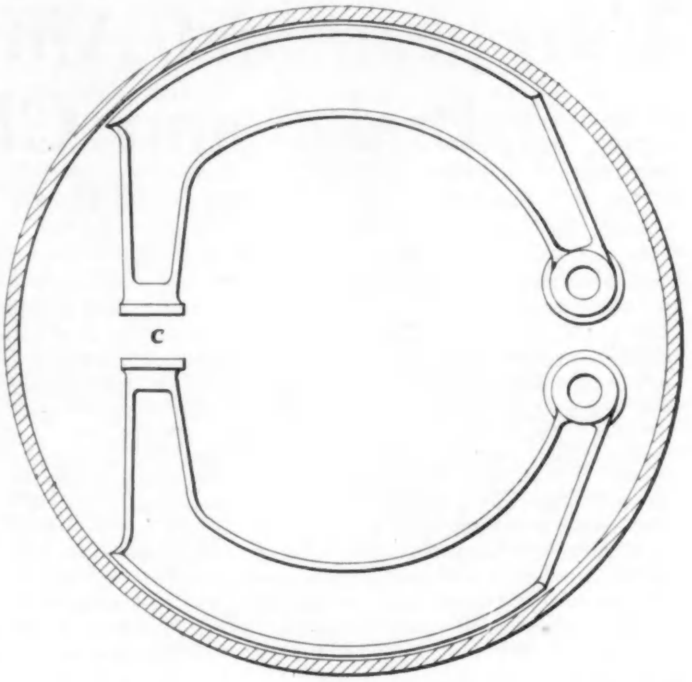


FIG. 3—A EUROPEAN INTERNAL BRAKE WITH TWO OPERATING PIVOTS

November, 1920. I have not checked these figures with those recently published; they are simply the result of my personal observation.

Seven nations were represented by 165 exhibits, Switzerland had 1; Holland, 1; Belgium, 3; Italy, 7; France, 28; the United States, 30; and England, 95. Of these 165 exhibits 49, or practically 30 per cent, were equipped with clutches of the single-plate type. There were 28, or 17 per cent, with clutches of the multiple-disc type. Only 4, or 2.5 per cent, had friction drives; that is, having the disc bearing on the ring, about 1 to $1\frac{1}{2}$ in. wide. This leaves 83 cars, or just over 50 per cent, that were equipped with cone clutches and with linings and leather, cotton or asbestos. Of the 95 English cars 56, or 59 per cent, had cone clutches. There were 11, or 12 per cent, with clutches of the multiple-disc pattern and 24, or 25 per cent, had clutches of the single-plate variety. The only friction-drive clutches of the 165 exhibits were of English manufacture.

Of the French cars 61 per cent had cone-type clutches, 18 per cent were of the plate and 18 per cent of the disc type. The only band clutch was of French make. Of the 30 American cars, 50 per cent were equipped with single-plate, 27 per cent with multiple-disc and 23 per cent with

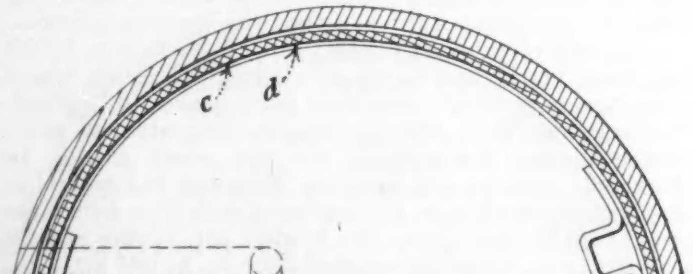


FIG. 2—ANOTHER INTERNAL BRAKE THAT IS OPERATED BY TWO LINKS

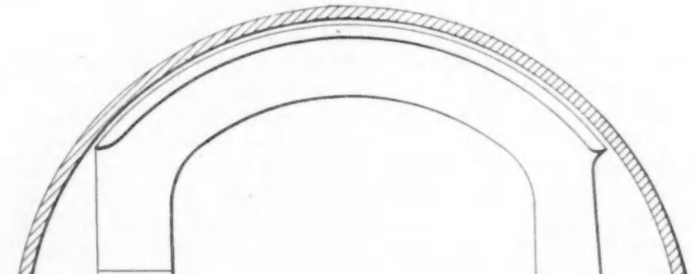

FIG. 4—IN THIS EUROPEAN BRAKE ONLY ONE PIVOT IS REQUIRED FOR OPERATION

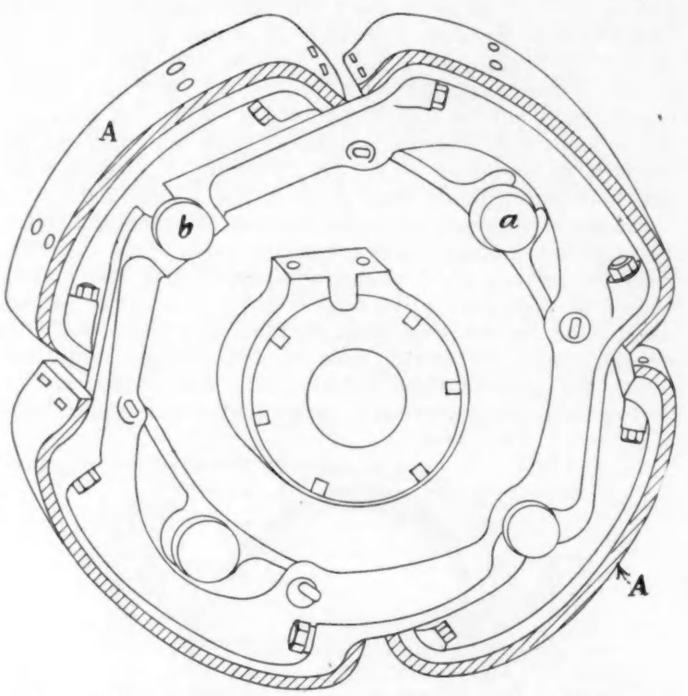


FIG. 5—A DOUBLE INTERNAL BRAKE IN WHICH NOT MORE THAN ONE-QUARTER OF THE DRUM CIRCUMFERENCE IS COVERED

cone-type clutches. With one or two exceptions, all the 30 American cars used the external brake for the service or foot-brake. Perhaps four or five used a transmission brake but, in general, the usual external and internal sets were used. With two or three exceptions, the English cars used internal brakes; most of them had transmission brakes also. The Continental cars stand about the same as the English with respect to brakes, but we often find the external brake used on the transmission.

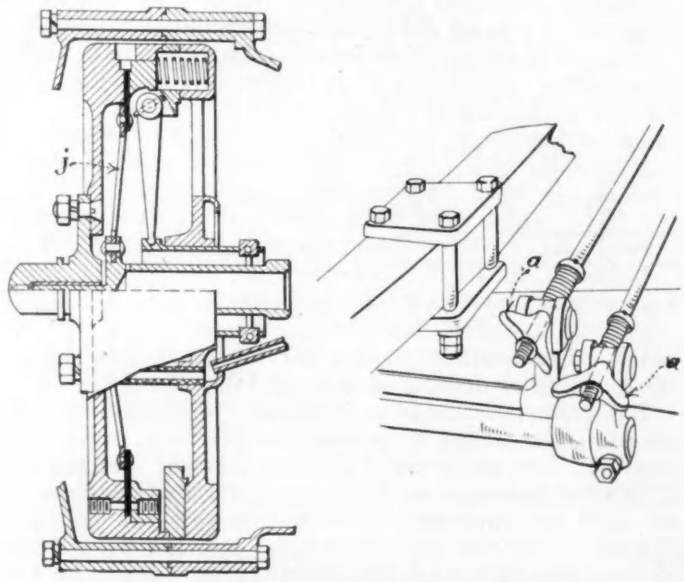


FIG. 6—TWO EXAMPLES OF EUROPEAN DESIGN
The View at the Left Shows a Clutch Having the Facing on Both Sides and the Springs at the Periphery While the Mechanism Employed for Operating the Brakes on a Belgian Car Is Illustrated at the Right

### CONTINENTAL COUNTRIES

The brake situation in Belgium is very similar to that in England; the internal expanding type is used almost universally and generally the brakes are metal-lined. There are cases where an asbestos liner is used and usually the brakes are fitted so that it can be used if so desired. Here also we find a considerable interest in the disc type of clutch. Of some eight or nine firms, there is a greater proportion that builds an engine that rates over 15 hp. than in England, although the actual number is really less. While those who use a cone clutch declare that they have no trouble with it, they are still a bit more than willing to try a clutch of the plate or multiple-disc type.

In Belgium we find one of the strong proponents of brakes on four wheels. The operating practice of one firm varies in a very interesting way from some others, as will be illustrated later. We find the internal brakes on the front-wheel drums usually operated by cams at the tops of the drums. This operating cam is on one end of the shaft that carries the lever and is fastened to the side-member of the frame. This shaft is furnished also with a flexible joint which allows for the steering movement of the wheels and carries the cam ring of the shaft. In the Belgian construction mentioned the brake is operated from the bottom and, by placing a specially shaped cam in the actual line of the steering-knuckle, carrying the operating shaft around with the wheel during the steering movement is avoided. I think there are only eight or nine cars built in Belgium, so that the field and the differences of practice are somewhat limited.

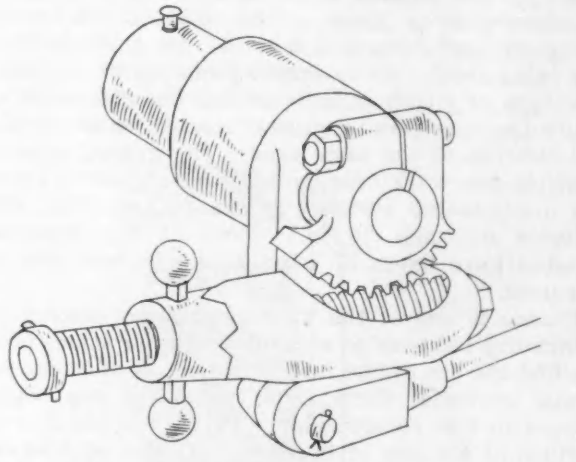

FIG. 7—A FORM OF BRAKE ADJUSTMENT IN WHICH THE ANGULARITY OF THE CAMSHAFT IS CHANGED

In Italy we find the internal brake and a rather larger proportional use of the transmission brake than in some of the other countries. The center of the industry in

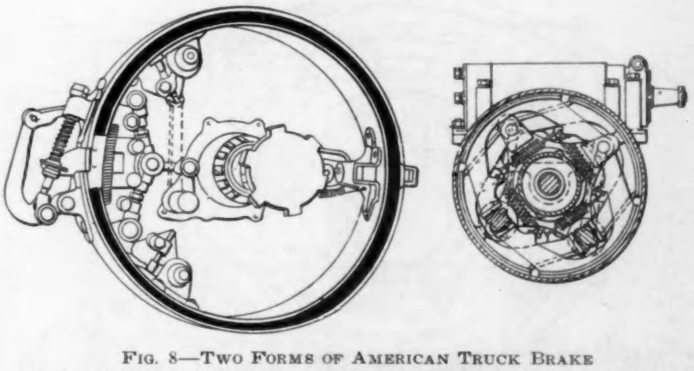

FIG. 8—TWO FORMS OF AMERICAN TRUCK BRAKE
At the Left Is a Simple Type of Toggle Brake and in the Brake at the Right a Certain Amount of Movement of the Cam Lever Is Necessary before the Lost Motion between the Shoe and the Drum Is Actually Taken Up

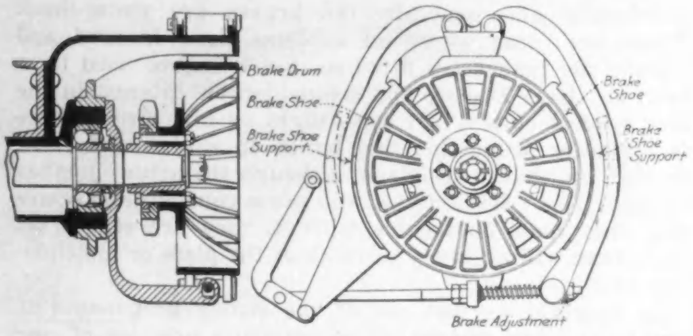


FIG. 9—AN EARLY FORM OF BRAKE IN WHICH AN EFFORT WAS MADE TO AIR-COOL THE DRUM

Italy is in the north, around Milan and Turin. Perhaps on account of the physical characteristics of the country, their engines are larger on the whole and their cars heavier and more rugged. Brakes are considerably more closely watched there than in England. Metal liners are generally in use. Brake-drums follow the other practice closely. We find brakes on four wheels in two or three cases. This feature is often made optional with the purchaser, at a price. The cone clutch is surely going out and more and more of the multiple-disc type are being used. An interesting feature of the multiple-disc type of clutch in Italy is that the diameters of the plates are considerably greater than those of the American clutches of the same type. The general practice in multiple-disc work here runs between 7 and 9 in., while the multiple-disc clutches of the Italian cars will run between $8\frac{1}{2}$ and 10 in. Most of the multiple-disc clutches have liners of some kind. In one case cotton was used.

France is one of the most interesting centers of the automotive industry as regards new applications of ideas. We find the use of the internal brake on the rear wheels almost universal there; and the drums are being increased in size considerably. There is a peculiar intermixture of the use of internal and external brakes; internal brakes on the rear wheels and an internal brake on the transmission, or an external brake on the transmission and internal brakes on the rear wheels. Some of the larger cars have brakes on four wheels. Usually they are listed as "extra." The matter of applying brakes and their operating connections to the steering-knuckles offers some peculiar difficulties. The one interesting application is on the Hispano. The brakes on the four wheels are operated by an auxiliary clutch that is brought into action by the foot clutch rather than by spring action. This auxiliary clutch operates on a shaft that is driven by a worm and gear from the transmission shaft. Considerably greater attention is given to the equalization of brakes, as a rule. Anyone who has used or driven a car equipped with brakes on its four wheels cannot help but be impressed by the ease with which a car is controlled and the quickness with which it can be brought to a sudden stop. By "sudden" I mean a stop that will bump one's head against the windshield.

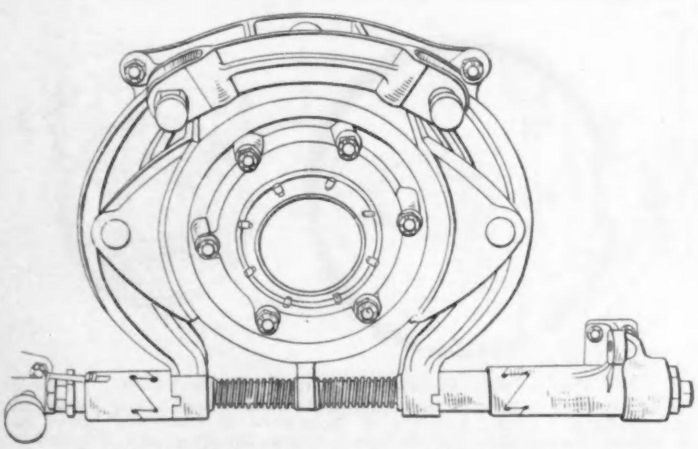

FIG. 10—THE PIERCE-ARROW BRAKE USING A RADIAL CAM

France still uses many cone clutches. Several multiple-disc and some single-plate types are used. One very interesting clutch is of the single-plate type; the clutch facing is required to do double duty, both sides being used. The facing is fastened to the metal plate that fits snugly inside the center hole of the facing. Another clutch that shows some ingenuity is one in which the shaft on which the driven member moves has been eliminated. The requisite motion for release is obtained by the buckling or dishing of the metal driven plate.

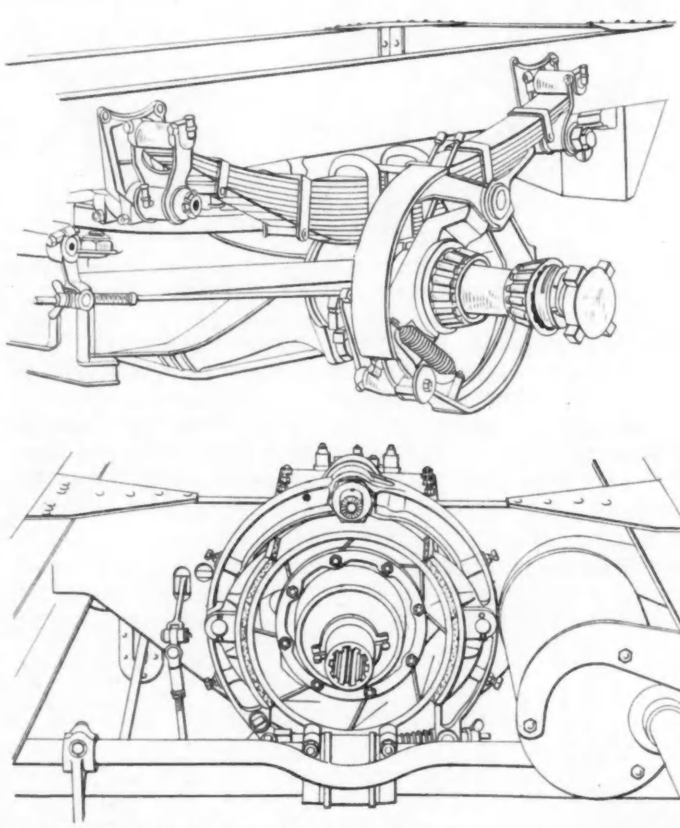

FIG. 11—TWO VIEWS OF THE BRAKE USED BY THE PACKARD COMPANY

European practice is so different from our own that we cannot criticise; it is based on economy, speed and comfort. The designers in France, England and Europe in general have just as much reason for their designs as we have for our own. French engineers tell me that French automobile body lines today are following those of American cars more than ever before. They also are following our practice of unit powerplant construction in some cases, but we find that the physical characteristics of the country and the gasoline price have a bearing upon European design and, when we criticise, we should take those factors into consideration.

## BRAKES

Fig. 1 shows a continuous-band brake. One of the great difficulties with this type is the excessive wear on the cam ends. The back usually becomes clogged with mud in spite of the fact that it is an internal brake, and

it often refuses to operate. Fig. 2 shows another brake operated with two links, and this practice is subject to the same trouble. The pressure at *a*, instead of being entirely tangent at *b*, is nearly radial. The wear is on the facing at *c* and there is practically no wear at *d*.

Fig. 3 shows one of the internal brakes used in Europe, and unusual in American practice; it differs in having the two pivots at *a* and *b*, with the usual cam-operating mechanism at *c* except that it is brought back from the circumference of the drum. Fig. 4 shows an internal brake of the same general type, except that it has only one pivot at the back. In two or three cases the brakes are operated by a wedge instead of a cam. Fig. 5 shows a double internal brake having only about one-quarter or less of the circumference of the drum covered. One set of brakes, *A*, is operated from the cam *a* and the other set from the cam *b*. The view at the left of Fig. 6 is a diagram of the clutch, in which the facing is used on both sides, with springs on the periphery. It depends simply on the buckling of the plate *j* to give the necessary motion for release. It is a De Dion clutch. Some of the mechanism for the operation of the front-wheel brakes on a Belgian car is shown at the right of Fig. 7. The cam *a* is in actual line with the steering-knuckle. The cam-operating shaft is brought back and operated from the front axle. The cam *a* is spherical, so that it allows the steering of the wheel without interfering in any way with the motion of the operating shaft. The Delage and one or two of the others operate their front-wheel brakes by a cam and a flexible connection from the frame. A universal-joint connects the cam with the operating shaft, the other end of which is held on the frame by a ball-and-socket joint. This construction allows the shaft to take any angle within the limits of the springs and still give no rotating motion to the cam.

In one of the English cars the thumb or wing nuts are placed at the back of the axle so that one has access to them without very much difficulty. Fig. 7 shows another one of the brake adjustments; it is a sector of a worm wheel and worm. The worm is turned by the ad-

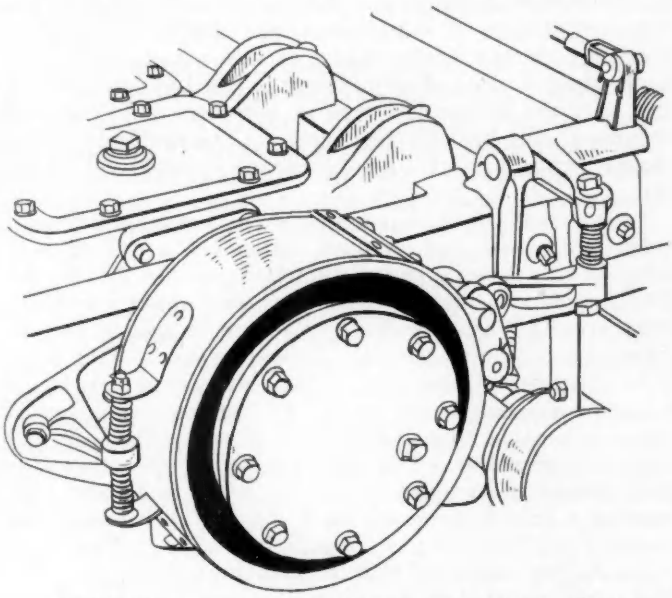

Fig. 12—Brake Employed on a Four-Wheel-Drive Truck

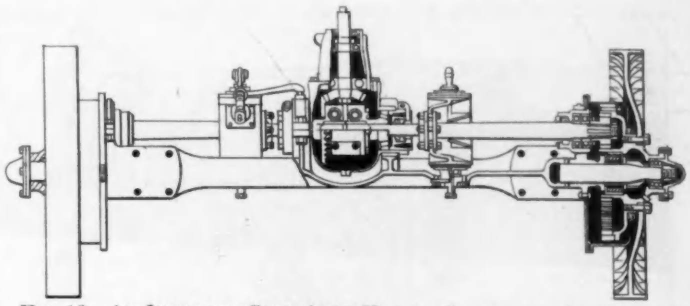

Fig. 13—An Internal-Gear Axle Having Superposed Auxiliary or Pinion Shafts

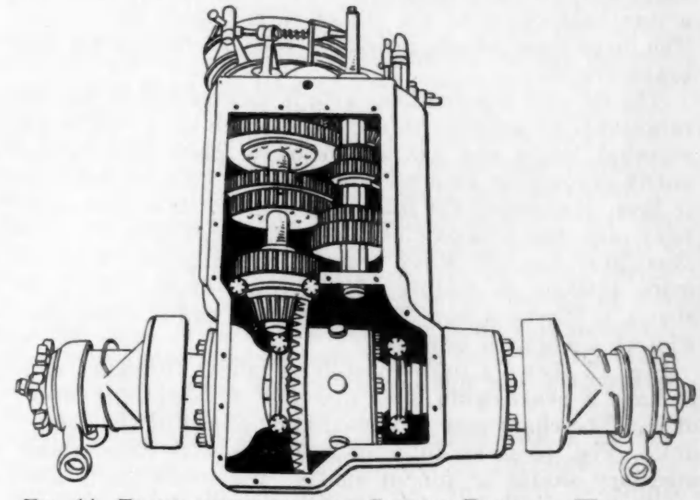

Fig. 14—Transmission for the Garford Truck in Which the Brake Band Is Located at the Forward End of the Countershaft

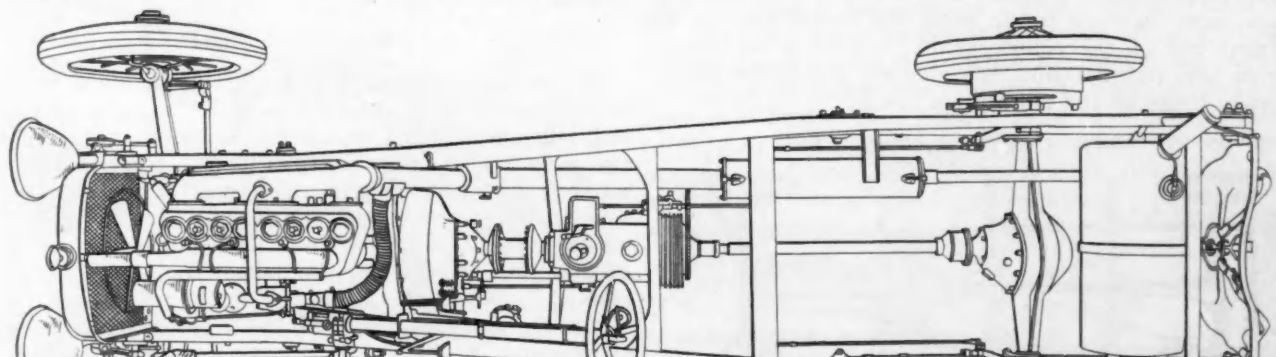

Fig. 15—A Passenger Car Chassis Having a Very Simple Brake Layout in Which the Brake Camshaft Passes through the Transmission Housing and Operates an Internal Cam on the Brake Back of the Transmission

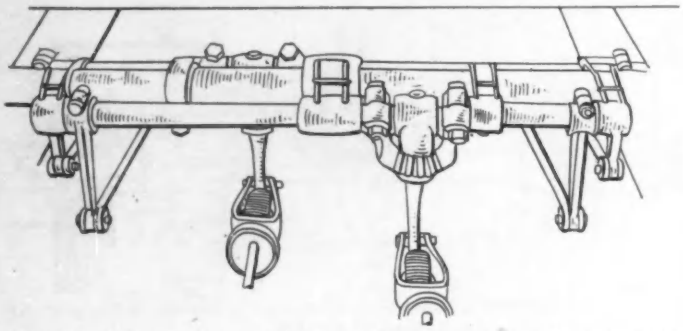

FIG. 16—A TRUCK-BRAKE CONTROL EQUIPPED WITH A DIFFERENTIAL EQUALIZER

justment *a*, drawing the rod along and so changing the angle of the camshaft *b*. The main objection to all these devices for changing the angularity of the cam is that, by changing the leverage, a point is reached where the wear of the brake is increased and this method is no longer effective.

Fig. 8 shows a simple type of toggle brake at the left and at the right is a Timken truck brake. In the latter it is interesting to note the position of the cams; there is a certain amount of movement of the cam lever before the cam actually takes up the lost motion between the shoe and the drum.

In an internal-gear axle of the Torbensen type one of the brakes is mounted on a pinion shaft. There is a cam action that operates the brake rather than the toggle. The large steel wheels provide a good anchorage for the brake-drums.

The Russell internal-gear axle is interesting in that it represents an attempt to enclose a brake-band. It is an external brake and has an enclosure plate that comes partly around the external brake-band. This is used also to keep dust out of the internal gears, but the same company also uses this axle on a passenger-car type. Fig. 9 illustrates the old Knox tractor, showing the attempt made toward air-cooling of the brake-drum. Fig. 10 shows a Pierce-Arrow brake, using a radial cam, and Fig. 11 a Packard brake.

Fig. 12 shows a four-wheel-drive truck. It seems odd to have a brake-drum upon one side with nothing back of it. The chain case is immediately ahead of the brake unit. Fig. 13 is an internal-gear axle with superposed auxiliary shafts or pinion shafts; the brake units are easily accessible and are mounted adjacent to the differential case. This same construction is used on some of the Kelly-Springfield trucks. Fig. 14 is a transmission from a large Garford truck, with the brake-band up at the front end of the countershaft. There is no direct drive on this transmission, the gearing and brake being similar to those on the old Mercer car.

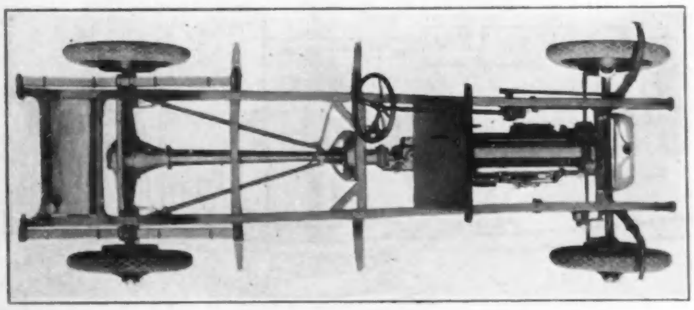

FIG. 17—PLAN VIEW OF THE DUESENBERG CAR IN WHICH THE INTERNAL BRAKES ACT ON ALL FOUR WHEELS AND ARE ACTUATED HYDRAULICALLY

Fig. 15 shows the Mercer chassis, which has a very simple brake layout. It has a brake camshaft that passes through the transmission housing and operates an internal cam on the brake back of the transmission. A wheel projects out from that shaft and the connection with the clutch pedal is made to the projecting lever by a small interconnecting link. A small thumb-screw at the top forms a very satisfactory adjustment. Fig. 16 illustrates the Atterbury brake-control unit, showing the differential type of equalizer and a method of keeping lubricant on the brake shaft; it has a very accessible hand-wheel brake-rod adjustment. The Lafayette equalizer is of the differential type on both sets of brakes and is immediately back of the gearbox. The designer of this chassis prides himself on not having a crooked rod on the car.

In the Panhard chassis a steel strip, having a series of holes in its rear end, is used instead of a rod or cam. This affords a very easy adjustment at the front end. The Lancia chassis for years has had the roller-chain links around the brake-drum to secure as nearly as possible an equal contraction all the way around. This principle is employed also on the Sunbeam car which has a pulley on the end of the lever. The brakes are operated through a cable that passes around one pulley, up and over another pulley. The cable running to the other side goes around the pulleys in the same way and back. By turning a thumb-screw all slack can be taken out of the brakes; this provides an equalizing effect. The Sunbeam builder developed this method through racing. Mr. Resta told me that it was found necessary to adjust the brakes during races and, to provide a scheme whereby the mechanic could do this quickly without getting out of the car, this method was the result.

The brake lever in the Delage car comes up in the center of the car and goes forward to the front brakes. The brake cam operates at the top and the shaft is flexibly mounted on the frame and universal-joint. The Renault company has brought out a differential brake-equalizer for two brakes on the rear wheels; as well as an equalizer in which there is a worm adjustment. The brake-shaft continues through and the clutch-tube floats on it. A sector of a bevel gear that merges with another bevel gear on the brake camshaft is mounted on one side. This brake camshaft is similar to that on the Mercer car. Fig. 17 shows a plan view of the Duesenberg car. This is another method of operating brakes, in the form of a flexible tube which has hydraulic actuation. These brakes are all of the internal type; they are four-wheel brakes.

## THE DISCUSSION

W. D. REESE:—Innumerable problems must be solved in connection with the production of a safe, economical and efficient form of motorbus. Among these problems the question of brake design is certainly the most formidable. Brake failures, irrespective of vehicle type, must be vigorously guarded against, but, of course, in such cases with a bus the potential hazard is much greater on account of the larger number of persons carried.

Mr. Farwell has intimated that comparatively few improvements in brake design have been made during the past decade and that while brakes are fairly efficient, improvement in design has not kept pace with the other units in the automobile. In a general way, Mr. Farwell's statements appear to be correct. At the same time, we believe that the design of brake employed on our buses is extremely satisfactory. But this does not mean

that we are unwilling to admit the possibility of improvement. As we see the situation, the fundamentals of good brake design from the standpoint of public service requirements are as follows:

(1) Safety
(2) Simplicity of adjustment
(3) Maximum service between adjustments
(4) Low upkeep-cost
(5) Freedom from loose parts and consequent rattle
(6) Ability to readily dissipate heat

During 1920 the Fifth Avenue Coach Co. carried approximately 50,000,000 passengers, equivalent roughly to half the population of the United States, and operated buses traveling 9,000,000 miles, which represents a daily mileage sufficient to encircle the earth. According to the statistics of our transportation department, this necessitated approximately 36,000,000 brake applications for passenger and traffic stops, or an average of four applications per mile. This does not include the applications made while running down grades, which would increase the total number of applications by several million. Approximately 10,000 ft. of fabric brake-lining supplied by various manufacturers was used during the year.

We have tested a very large number of different brakes in various ways and excellent results have been obtained from those now standardized on our Model A bus. For example, tests were made with buses weighted with sand-bags to the equivalent of a full passenger load to determine the maximum braking that could be obtained with normal effort on the part of the driver. Many tests were conducted on Broadway, New York City, between 136th and 150th Streets, making runs in each direction with dry road-surface conditions and accurate data were arrived at by a recording device consisting essentially of four electromagnetically operated pointers, a time-marker clock and a contact-making device mounted on the hub on one of the rear wheels of the bus. One of the pointers was actuated by the time-marker clock to indicate 1-sec. intervals, another by the contact-making device on the wheel to indicate the number of revolutions and two other pointers by push buttons to indicate the length of the braking period on each brake. All stops were made without skidding the rear wheels. The results of a large number of tests showed that with normal effort on the part of the driver the deceleration obtainable was 3.75 m.p.h. per sec.

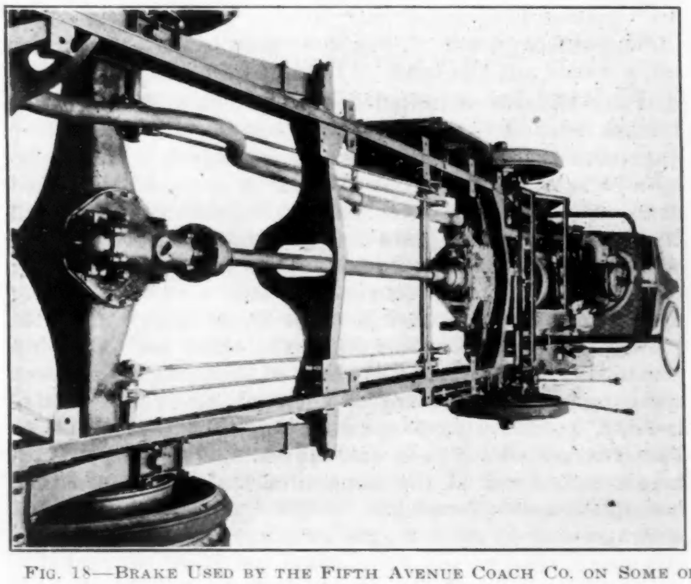

Fig. 18—Brake Used by the Fifth Avenue Coach Co. on Some of Its Buses

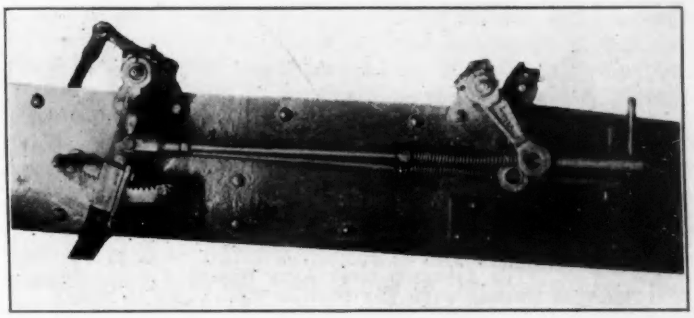

Fig. 19—A Type of Brake Adjustment That Eliminates the Use of Turnbuckles

Fig. 18 shows the brakes on the Model A bus. The brake-pedal is operated in a conventional fashion by the right foot and the hand-brake by the right hand. Pressure applied to these members is converted by suitable linkage through a pull on the rods leading back along the side of the frame to the cross-shaft. At this point there are hooked up four rods running to the cam-actuating levers on either side. No equalizers are used since they are unnecessary when only one point is made use of for service adjustment.

It will be realized readily that in bus work, especially with a Hotchkiss type of drive, a rather difficult problem confronts the engineer who attempts to design a brake-operating mechanism, especially when we consider the tremendous deflection and consequent axle movement that are necessary if we are to have a vehicle that rides with the minimum amount of discomfort to the passengers and at the some time assures a perfect-acting brake under either the full or unloaded condition. To take care of the deflection we use a center cross-shaft which permits of a comparatively long rod to the cam operating lever. The position of the cross-shaft and the length of the levers used have been determined as being the best combination of theory and practice obtainable after a vast amount of experimental work. To eliminate spring trouble we find that it is necessary to test all of our springs on a spring-testing machine at regular intervals. It will be appreciated that one cannot get a perfect brake action with a weak spring on one side and a comparatively stiff spring on the other.

The actual wear on the band is taken care of by setting the levers, which are placed on serrated shafts throughout the entire mechanism, and also by shims which are placed on the top of the brake anchor-plate. The brake-band is made up of a single piece of spring steel to prevent deformation. There are no joints and the ends are made perfectly symmetrical so that they can be turned upside down when the upper part is worn, this being of course the first part to wear in the wrap-up type of brake. The band is hung on a spider attached to the axle and is perfectly free to rotate, its movement being limited only by the cam. A 20-in. diameter is used with a $2\frac{1}{2}$-in. width, which gives a total braking area of about 600 sq. in.

The cam used for actuating the brake is flat on top and has a radius on the bottom such that the movement of the brake-band anchor is just proportional to the pedal or lever travel. As the cam nose moves downward, it drives the band against the drum and the remainder of the braking action is accomplished by the dragging effect of the lining, which tends to intensify progressively the pressure around the surface of the band. This is proved by the fact that the greatest amount of wear comes at a point about 6 in. back from the upper brake-band anchor-plate.

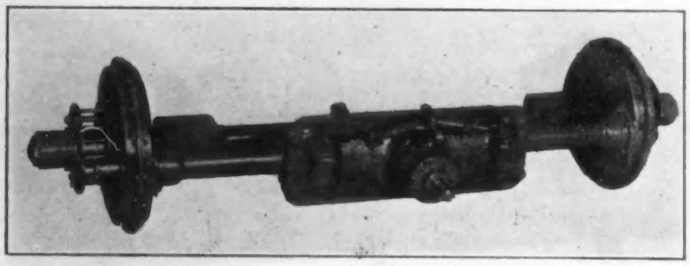

FIG. 20—A 1-TON INTERNAL-GEAR AXLE HAVING A SOLID STEEL CARRYING MEMBER WITH THE DRIVING MECHANISM IN FRONT

The drum we are using at the present time is of a special-alloy cast-steel. We have made extensive tests with pressed-steel drums, but these have always proved unsatisfactory. At present we are experimenting with a heat-treated forged-steel brake-drum having a high carbon-content, which has uptodate given extremely satisfactory service.

It is interesting to note in passing that the hand-brake is arranged so that it pushes forward for application, which is just the reverse of conventional practice. The object of working the lever in this manner is that the hand has a shorter distance to travel for starting braking than it would have if one had to reach for the handle and then pull it back. This saving in time is often enough to avert a serious accident.

Fig. 19 shows a type of adjustment that eliminates the use of turnbuckles and permits road adjustments to be made rapidly and without getting under the vehicle. The tube shown, which acts as a nut, takes the ends of the rod and shortens or lengthens it as desired. It is designed so that the threads cannot possibly be damaged through carelessness. The pin, which is used as a lever, makes the tube unbalanced and consequently has no tendency to turn or change its adjustment through vibration.

The major portion of the brake adjustment is made in the garage after every 2000 miles of operation. At this time the wear on the lining is compensated for by shimming up under the brake anchor-plate so that the distance between the cam and this point of contact on the brake anchor-plate remains approximately the same throughout the life of the lining.

R. W. HASTINGS:—The firm I represent was organized to produce an improved truck-axle. To accomplish this the three factors selected for improvement were (*a*) increased accessibility, (*b*) proper enclosure and lubrication and (*c*) adequate brakes. In considering the most

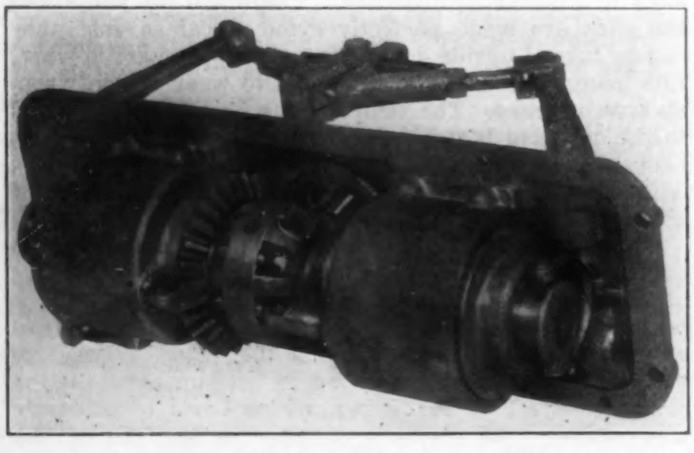

FIG. 21—THE APPLICATION OF THE BRAKE TO THE AXLE SHOWN IN FIG. 20 IS IN TWO SIMILAR UNITS, ONE ON EITHER SIDE OF DIFFERENTIAL

important features in the design of the axle, we have defined the term "adequate," as applied to our brakes, to mean a brake of large capacity, designed to deliver dependably uniform service without replacement for a period equal to the average life of the vehicle itself. Such a brake as compared with one of the band or shoe type must present an opportunity for greatly increased frictional area, an evenly distributed pressure to utilize this area fully and a complete enclosure of the mechanism as a protection from the abrasion and unreliability resulting from the introduction of foreign matter. The disc or clutch type of brake seems to fulfill these conditions best. It can be enclosed readily and, when properly designed, seems to possess qualities making it almost indestructible.

Fig. 20 shows our standard 1-ton internal-geared axle, with the usual solid-steel carrying member and the driving mechanism in front of that member. Our internal gear is enclosed in an oil-tight case, just inside of the wheel. The pinion has jaw engagement with the drive-shaft, providing for the removal of the drive-shaft without disturbing the gears, the wheel hub and bearings or the case surrounding them.

The application of the brake, Fig. 21, to our axle has been accomplished within the enlarged differential housing, it being applied in two similar units, one of which is located on either side of the differential and attached directly to the drive-shaft. The construction of the brake parallels closely that of the multiple-disc clutch, Fig. 22. A set of stationary plates of molded asbestos is slidably held within a housing that is secured to the front cover-plate. Steel rotating plates of special double construction are placed alternately with these friction plates

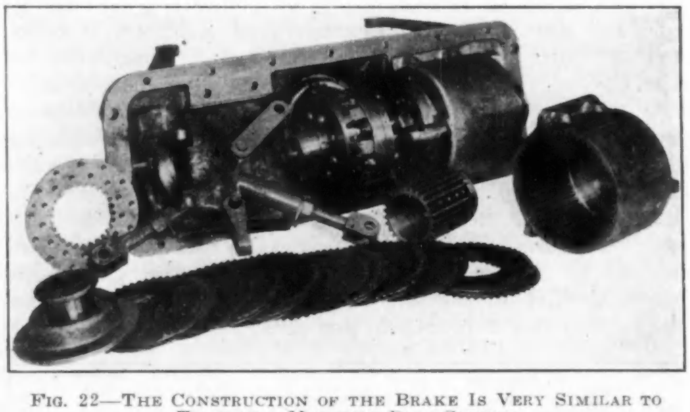

FIG. 22—THE CONSTRUCTION OF THE BRAKE IS VERY SIMILAR TO THAT OF A MULTIPLE-DISC CLUTCH

and are slidably mounted upon the hub member, which rotates with the drive-shaft by virtue of a splined engagement thereto. End-pressure is applied to the plates by a bell-shaped pressure-plate that is actuated by forked arms attached to two vertical shafts extending through the top of the cover-plate and terminating in lever arms which carry the toggle equalizing members. The toggle mechanism, Fig. 21, consists of two members having cam-shaped ends so that as their inner ends are pulled forward, thus separating the lever arms and applying the brake, the point of contact at the center between these cams does not travel forward but remains stationary, maintaining a constant angle of toggle action. This feature allows us to take advantage of the powerful toggle action and at the same time maintain a constant multiplication of leverage. While the multiplication of leverage due to this toggle construction increases the effective pressure on the brake, an item of probably greater interest and value is the automatic equalizing of

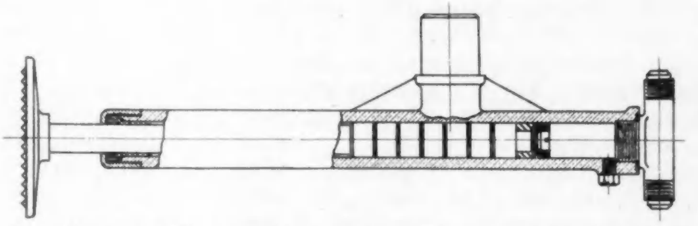

Fig. 23—Pedal Used on Pierce-Arrow Equipment in Connection with a Hydraulic Brake

the brake pressure thus accomplished. With the brake released, the toggle members are held by the return spring against a locating seat provided on the face of the cover, thus maintaining proper and equal clearance or opening for each of the brake units. Depression of the pedal draws the toggle forward, releasing it from this locating seat and thus leaving the entire system of levers free to swing to either side and so compensate for uneven adjustment. With the mechanism in this free position, it is evident that the reaction from the pressure upon one brake unit finds no resistance except that

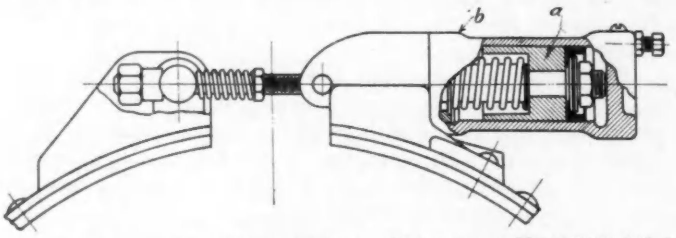


Fig. 24—Construction of the Band Mechanism Employed on a Hydraulic Brake

of the pressure upon the other unit, for which reason an absolutely even pressure upon each brake is guaranteed.

The question of oil circulation has been given considerable study. We have provided a means for introducing oil at the center of the brake that allow it to flow out through the specially spaced rotating plate, thus carrying away the heat that develops in the brake and giving it ample chance to radiate from the large surface of the axle housing. This provides an unusually cool brake and we have found almost no condition under which it is not possible to place one's hand upon the axle.

Proper adjustment of the brakes is made by releasing the outer nut on the toggle cams, following this with a similar manipulation of the inner nut, setting the brake arms over and moving in the pressure plate. Enough clearance is allowed to wear out the brake without any other adjustment. We have run one job about 19 months and still have the same plates; they show almost no sign of wear.

A Member:—I will relate my experience with the brakes on the Delage car from a sales standpoint. There are various advantageous factors about using brakes on all four wheels. One of the first and most important is comfort. We find that, no matter how suddenly the brakes are applied on all four wheels, there is less tendency to throw the passengers forward and out of the seats. Instead, we find a tendency of the entire chassis to sink into the wheels. In fact, one can see the hub cap sink an inch or two, as the brakes are applied harder. Another very important feature of four-wheel brakes lies in the increased safety they afford. These brakes can be applied on wet days, in snow and on ice, without chains, just the same as would be done on a dry pavement. The effect seems to be the same, with the possible exception that the brakes sometimes cause all four wheels to slide. But during an experience of 18 months they have never caused side-skidding. The general economy on brake-linings is another important item. There seems not to be the same amount of jar, because the brakes are seldom put on hard. A very slight pressure of the foot will stop the car. The economy on tires is very marked, due probably to the fact that the rear wheels do not drag. The tires seem to give much greater mileage than when using brakes on two wheels only. We have never determined the increase in mileage, but it is surprisingly large. I do not know whether that is due to the brakes or to the

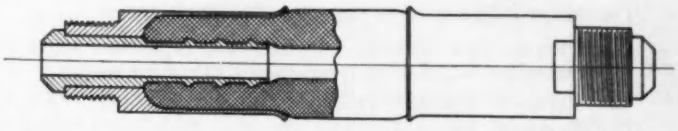

Fig. 25—Special Hydraulic Hose and a Somewhat Unusual Fitting Are Employed to Overcome Motion between the Frame and the Axle

car, but I feel that the brakes are responsible largely. These four-wheel brakes make it very easy to drive safely in traffic. One can run up closer to the car ahead. Last and most important from a sales standpoint, we find that when a man acquires the habit of driving a car with four-wheel brakes, he is less inclined to buy one having less braking power.

On the Delage car, we find that the brakes fulfill all our requirements under all conditions. The adjustments are very simple. There seems to be no wear. With four brakes, we have double the braking surface, with half the braking effort. Altogether, we find that four-wheel brakes afford very comfortable riding and are very

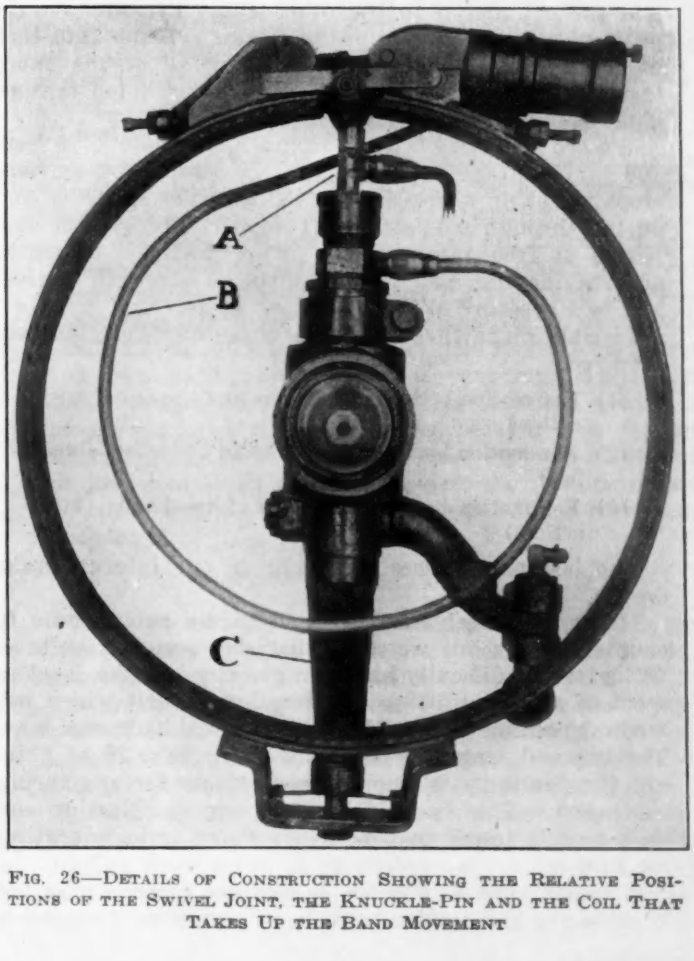


Fig. 26—Details of Construction Showing the Relative Positions of the Swivel Joint, the Knuckle-Pin and the Coil That Takes Up the Band Movement

much favored by the public. The people who have driven cars equipped with the four-wheel brake in Europe are very enthusiastic regarding their operation.

MONTGOMERY MAZE:—The four-wheel hydraulic brake that we are now manufacturing can be considered only in the light of an accessory. Whatever we accomplish in the way of replacement equipment can be viewed only from that standpoint, for it has been necessary to work around existing conditions of design which are far from being uniform or desirable from our point of view. Future factory equipment design can readily excel in looks, efficiency and cost.

The main items increasing our efficiency are

(1) Perfect equalization, automatically obtained since the application is through fluid pressure only
(2) Complete freedom from mechanical linkages
(3) Complete absence of any effect on braking arising from relative frame and axle movement
(4) Use of 100 per cent of the car weight as a source of road friction instead of a fraction of the normal weight on the rear wheels

All four wheels are operated simultaneously by a single pedal in the conventional manner and the degree of retardation is entirely within the control of the operator. The system is purely a displacement proposition, and an emergency stop can be obtained only by the application of the emergency pressure.

Fig. 23 shows the pedal we use for Pierce-Arrow equipment; on the Cadillac the standard pedal is not disturbed. It has a normal travel of approximately 3 in., the reserve being sufficient to wear out a third of the lining without adjustment.

Fig. 24 illustrates the construction of the band mechanism. The piston *a* is attached to one end of the band and the cylinder *b* to the other end. Pressure on the pedal displaces liquid from the master cylinder into each of the wheel cylinders alike, drawing all of the bands together with an equal force. It will be noted that no band can grip until all are in contact.

We are operating with a normal line-pressure of from 100 to 150 lb. per sq. in., but it is possible on extreme stops to set up a pressure of 750 lb. The liquid is conducted through a special, soft copper tubing, well supported at frequent intervals. This tubing is drawn to our specification and has an ultimate strength equivalent to a pressure of 13,800 lb. per sq. in.

In addition to this rigid line, three compensations are required

(1) For motion between the frame and the axle (spring action)
(2) For motion between the axle and the wheel (steering)
(3) For motion between the ends of the bands (brake action.)

The last named does not occur in our internal-brake design.

It has long been our intention to use rubber hose to counteract (1) but we were unable to secure a suitable fitting. This difficulty has been overcome by the development of a special fitting of exceptional merit which has made it possible to use a length of special hydraulic hose. The exposed length of hose shown in Fig. 25 is 1 in. and the flexibility is ample to compensate for any spring movement. This hose is built to our specification and each unit is tested to 3500-lb. pressure under vibrating conditions.

A swivel joint *A*, Fig. 26, is mounted directly over the knuckle-pin and in line with it to permit perfect steering. Braking cannot affect steering in any way, nor can steering affect braking. A coil pipe *B*, 40 in. long, takes up the band movement on the external brakes. This movement has a maximum of 3/8 in. The method of mounting the front-wheel equipment on an Elliott type axle is shown in Fig. 26. The swivel is mounted on the knuckles and the knuckle-pin *C*, being stationary, carries the lower half of the band. On the reversed-Elliott type of axle, the swivel is mounted on the knuckle-pin and the lower half of the band is supported from the steering arm which we replace in this case.

There are three points of closure in our system; (*a*) the copper-tube connections which are S.A.E. Standard flared-tube fittings, which are satisfactory in every way (we have never had a leak at this point); (*b*) the swivel packing boxes (these hold a maximum of special packing; because of their design that permits the operating pressures to compress the packing further, we have yet to encounter a single failure); and (*c*) the cylinder cup leathers. The last named has been a source of great difficulty but we have located a cup that has failed to show any leak during a 4-year test; with the stabilizing of the leather market, we are now able to secure a uniformity of material that obviates any further leaks at this point.

To make our system complete, however, we have adopted a standard reserve tank; it will be necessary every few months, due to slight seepage, to draw liquid from this tank. By keeping the tank full and filling the line only in this manner, all chance of drawing air into the line is done away with.

Brakes are uppermost in the mind of any owner in hilly country and he grasps at any remedy for his constant worry; but, regardless of experience, it takes but a single demonstration under any conditions to impress upon the operator that brakes have at last reached the plane of present-day engine design.

In addition to the safety factor given by the brakes, the owner also gets an actual monetary return. Tires can be made to show a 30-per cent increase in mileage from the standpoint of tread wear. Adjustments are not required at less than 10,000 miles, and relining of brakes is unnecessary under 30,000 miles. These figures are an average obtained from our test-cars; it is very doubtful if any owner will ever subject his car to the extreme and continued operation that they received.

HERBERT CHASE:—Brakes must absorb power quickly when they are applied but at other times they ought not to absorb power. One common fault with both American and foreign-built cars is that brakes do drag more or less. Possibly the external-band brake drags more than the internal. There is room for better construction in brakes in general. The average brake does not compare well with the other parts of the car in the quality of its construction. British criticism of the American car with respect to the brake construction is rather caustic and, in some cases, probably is justified. It should be borne in mind, however, that British cars cost more than American cars on an average, and more expense can be put into their brake construction.

What conclusion has Mr. Farwell reached about the metallic brake-lining that is generally used abroad? There are several different kinds, I believe. I wish to know how they compare with the fabric generally used here in regard to the value of the friction coefficient. Also, will Mr. Carson describe the pressed lining that I understand he has been working with? Possibly he will describe also the testing apparatus that is being developed by the Bureau of Standards for determining the

relative merits, including the wearing qualities, of different linings.

H. G. FARWELL:—With reference to the relative coefficient of friction of the metal brake-lining and the fabric lining, there seems to be considerable discussion and difference of opinion. The coefficient of friction will run approximately 0.4; with a bronze shoe it will run approximately 0.2. That has been corroborated by engineers from abroad. They use more encased drums abroad than we do here; pressed-steel drums are used almost altogether.

V. W. PAGÉ:—I witnessed some tests of the multiple-disc brake. I was afraid there would be considerable drag and attendant heating. After a number of tests down a steep test-hill, I was able to put my hands on that casing without any discomfort; a standard touring car that accompanied the test car stuck on this same test. I could not place my hand anywhere near the brake-drum on account of the heat. Then we tried some coasting tests. We find fully as good results with that form of multiple-disc brake as would be obtained with the conventional band-brake well adjusted, and considerably better results that one would get with a band-brake ordinarily adjusted.

C. CARSON:—Mr. Chase has requested information regarding the testing apparatus at the Bureau of Standards. The Parts and Fittings Division of the Society's Standards Committee was assigned the subject of brake-lining and it undertook to develop some standard method of testing brake-linings in collaboration with the Bureau of Standards at Washington. The results so far obtained have not entirely solved the problem, but very gratifying progress has been made.

A pressed-steel drum was mounted on the shaft of a dynamometer and a skeleton frame of the general form of a prony brake was built around it. A spring-balance was installed between the arms, to adjust the load on the brake-shoes. Two short flexible bands were used instead of a complete encircling band, as is commonly found on wheel brakes. These carried linings about 11 in. long, 2 in. wide and 1/4 in. thick, and the pressure was applied so that a nearly uniform pressure per square inch would result. First, we tried to find what pressure per square inch could be carried and what velocity should be used. To accelerate the test, an attempt was made to run the apparatus with a water-cooled drum. That was provided by putting a plate equipped with the usual tube for introducing water in the open end of the pressed-steel drum. But when using high pressures and high velocities, heat is generated so much faster at the point of contact with the lining on the drum than it can be transmitted through a steel drum 1/4 in. thick that, even with a drum containing water, the surface of the drum will fuse while it is running. So, we were forced to abandon the theory of running at high pressures with a water-cooled drum, because the water dissipated only a limited amount of the heat generated. The tests are not yet completed. We have made a long series of tests using both water-cooled and dry drums. Apparently, a character of lining that will give excellent results at moderate loads will break down and give very unsatisfactory results when high pressures and continued brake applications are given to it.

It seems unfortunate that as yet, all through the industry, there has been no standard method of testing such materials. Some firms have tested brake-lining by making what might be called a skid test. They apply a piece of friction material to a rotating drum, hang a certain weight on it, make it turn for a certain number of hours at a certain number of revolutions per minute and record the result. The material that endured the longest was given the credit for being the best, without taking into consideration the power absorbed by the brake during the run. We have tried to eliminate such a condition in the apparatus we use. At present, the doing of a uniform amount of work is taken as a basis for the test. The pressures per square inch are varied to make the power consumption constant at all times. We keep the revolutions of the dynamometer and the power consumption constant and change the pressure on the lining. We feel that, if the linings are doing the same amount of work, we can then approximate a fair comparison of their life.

Among other interesting results, we found that some of the yarn in the linings had been made with brass-wire cores and that the surface of the lining became covered with copper plating; to a certain extent the steel drum was coated likewise. Investigation indicated that the heat generated was sufficient to drive the zinc out in the form of vapor or dust. We have found that the degree of vulcanizing in the rubberized linings seems to have a very pronounced effect on their wearing quality. Linings made from the same fabric and having practically the same rubber mixture, but different degrees of vulcanizing, will vary in their performance under the same load conditions from 20 min. for one sample to 16 hr. for another.

A recommendation probably will be made to the users of brake-linings, asking them to modify, if possible, their method of installation. A prevalent method of installing lining requires the strip to be very flexible as the usual practice is to rivet it at the ends first and then press out the kink left in the center to make the lining hug the band tightly. That requires a considerable flexibility in the material. We believe that in adhering to that flexibility the users of lining are sacrificing much of the life of the lining. That is indicated by the change in wear according to the degree of vulcanization. With a vulcanized or with the woven type of lining, hard pressed and impregnated with a hard compound, this kinking method of installation could not be used and there would have to be a change in the method of application.

We find that they are using very hard cured lining in European practice. The Ferodo lining, which has a corrugated shim between the lining material and the band to circulate air and dissipate the heat, is an example of such material. This lining is an asbestos woven fabric; it is very hard, compressed to a very high degree and almost lacking in flexibility. It is installed usually in comparatively short curved pieces, because it is not flexible enough to bend around a band. My opinion is that, for long life and maximum service, the present method of installing brake-lining and its degree of hardness must be changed.

### COEFFICIENT OF FRICTION OF BRAKES

The determination of the coefficient of friction is perhaps the most elusive problem we deal with. We have not even been able to determine it as a constant on any one particular sample during a run. We doubt very much if a really fixed coefficient of friction can be maintained with an impregnated, woven, or folded and stitched fabric, or for any fabric composed of yarns interlaced. It may be reached in some new form, such as unwoven or pulp lining commonly called molded material, used in some types of clutch-facing. A rough value for this coefficient would be about 0.40, but I think possibly that it should be modified to 0.36 for a woven and 0.42

or 0.43 for a rubberized lining, the latter having a slightly higher coefficient. Almost any coefficient desired can be obtained by changing the compound. One can make a lining having a severe grip or, changing the compound by introducing certain other ingredients such as waxes in one form or another, secure a low coefficient.

From the tests, we believe that the reason for the variation in the friction coefficient is that at no two times during the wear of a piece of lining is there the same condition of surface contact. Consider a piece of folded and stitched, laminated lining with a rubber compound. At the beginning, there is a veneer or surface of rubber that has a certain coefficient of friction on the steel drum. As the wear progresses, the coefficient changes because the surface contact is composed of a certain area of asbestos fiber, metal and rubber. The areas of the three materials in contact change continuously and the coefficient of friction changes correspondingly. While there is no uniform cycle of performance, there is a fluctuation and continual variation, even in the same piece of lining.

L. G. NILSON:—Who has had any experience with the metallic brake-lining that is a composition of lead and copper?

MR. CARSON:—We have not tested any of that material. I saw some clutch-rings that were made of that material recently. The engineer who conducted the test of the rings eliminated that material on account of its high cost and because the coefficient of friction was so low that it would have been necessary to increase the area of the clutch to a prohibitive amount to use it interchangeably with asbestos materials.

W. C. MARSHALL:—What tests have been made to show which type of brake is freer from oil, the internal or the external? The efficiency of the brake depends largely on whether the oil gets in on the brake-band and the drum. In some cases the oil might be thrown off. In other cases it might hold.

MR. FARWELL:—Our experience shows that oil gets in on both types. Probably a greater amount of the oil will be retained on an internal than on an external type. We find, in some cases, oil or grease on both the internal and external bands. It seems to involve choosing the lesser of two evils.

N. G. BERGENHOLTZ:—In regard to the brakes on the buses of the Fifth Avenue Coach Co., it seems that many of these are not shoe brakes and do not act equally in either direction. On this particular brake, our attention was called to the fact that the cam was not of the same shape on both sides. How does that brake act in going in a reverse direction? Does it give an equal braking effect?

MR. REESE:—No, it does not. The brake that we use is known as the "wrap-up" type. The efficiency is the greatest when going in the forward direction. When going in the reverse direction, it is much harder to apply. I should judge that if one could apply the necessary pressure the efficiency would be the same.

MR. BERGENHOLTZ:—Is the object of that cam shape only to take up the slack first, before the pressure is applied, rather than to try to equalize the pressure in both directions?

MR. REESE:—Yes.

M. C. HORINE:—A self-wrap brake is essentially a one-direction brake. Brakes have been developed which are known as the double-wrap type but a double-wrap brake, either external or internal, is practically not a wrap-type brake at all. That is, it is not a snubber; it does not work on the principle utilized when several turns of rope are taken around a capstan, as is the case with the ordinary self-wrap brake, because the self-wrapping on one side is compensated for by the unwrapping of the other. This matter of having a brake act equally in either direction is important, particularly in motor-truck work; great weights must be considered and gear-changes are not so certain on a grade, because the truck is going at a slower speed and its inertia will carry it forward a very much shorter distance. The experience of the company I represent was such that, previous to the time that type of brake was abandoned, it was necessary to redesign the cam so that it had equal action on both ends of the band. The only advantage of the flexible band in a double-wrap brake is to give a more or less equalized pressure.

The effects of pressure and speed on the wear of a brake have been suggested. The amount of wear on a brake should be roughly commensurate with the amount of energy dissipated. It seems that a brake might be designed with a small drum operating under very high speeds. This would mean lower friction at higher speeds and no more wear per square inch than with a larger brake operating at lower speeds at higher pressures. It seems to me that the wear would be dependent upon the amount of area. There is considerable buncombe with regard to braking area. It is possible to design a brake with a great braking area, much of which is worse than parasitic. Taking the shoe illustrated in Fig. 27 as an example, it would be possible to line it up to the tip where the cam contact is and down to the hinge. Such a brake acts as a lever. The portion of the shoe at *a* is on the wrong end of the lever. One can get very little pressure at that point, whereas at *b*, close to the fulcrum, one can get a great amount of pressure. If the portion *a* is lined, the lining will act as a spacer. It is impossible to get sufficient pressure on it to make an effective brake, and yet it acts as a spacer to prevent the portion *c d* of the lining which produces effective braking, from making contact. The portion of the lining at the point *b* does not reach the drum after a certain amount of wear, because it is so close to the hinge; so, it is largely parasitic area. On a rigid drum of this character I believe it is not necessary to have a lining for more than about the distance *c d*. I do not believe that any more lining on that shoe will give braking effect. In a case where it is close to the tip, it may prevent effective braking.

In regard to having internal and external brakes on the same drum, which is the conventional practice on touring cars and the cheaper kinds of truck axle, it does not seem right to me to put two brakes of any type on the same drum. Asbestos is hardly a good conductor of heat, whereas iron is a very good conductor. Since the drum is the brake member that contains most of the heat, it seems reasonable that the binder in the lining should burn because the drum gets hot. If one could always apply a brake to a cool drum the lining would not burn. Suppose we have two sets of brakes which we apply alternately in descending grades to avoid burning either set. If we apply the alternate sets of brakes to the same drum, which has already become heated by the application of the preceding set, the second set will burn almost immediately. Hence, the ideal brake arrangement would be for each set of brake shoes to act on a separate drum, which is the condition we have with four-wheel brakes and with shaft brakes.

Another consideration in connection with shaft brakes is that of equalization. It is possible to equalize the pressure on two brakes in a number of ways, such as using a simple cross-tree, a differential arrangement,

pulleys, fluids or other means. But that only equalizes the pressure and does not equalize the braking effect. It appears to me that the only way to equalize the braking effect is through the differential, inasmuch as two tires will never follow exactly similar tracks and it is the tire on the road that actually gives the braking effect. That seems to me a strong argument in favor of the shaft brake. However, the shaft brake has some defects which I think have not been given sufficient attention.

The greatest complaint against shaft brakes is that they chatter. Chatter in the shaft brake is due to many causes, chief among which is the fact that the brake itself is not supported firmly enough. Most shaft brakes overhang on a propeller-shaft and, naturally, since there is no propeller-shaft that remains concentric, there is a slight wobbling of the drum which ordinarily is supported separately from the shoes. Another cause of chatter is the looseness with which the actuating means and the shoes themselves are attached. If the shaft brake is properly designed with rigid shoes and a drum that is mounted between bearings, as one would hang a grindstone, and if the entire brake and its mechanism is supported by one rigid frame, there will be no chatter. I know this because I have experimented with brakes of that character.

The ordinary method of mounting a shaft brake is to have it operated by a pedal. A number of cars now obsolete had brakes of that sort, and it was characteristic of the operators of those cars that they almost never used the foot-brake. The operator always used the hand-brake because the shaft chattered. If it must chatter, the shaft brake should be operated by the hand-lever, because ordinarily the hand-lever is the one which is used to lock the car when parked. This means that it is generally applied when the car is stationary. The foot-brake ordinarily is used only when the car is moving; therefore, if we must have a chattering brake, let it be the hand and not the foot-brake. Another reason why the shaft brake should be operated by hand is that spring action does have the effect of shortening and lengthening brake-rods that are connected to the rear axle, and that it is a very common experience when the brake is applied with the car loaded to have it release itself when the car load is taken off. With very light cars, where the hand-brake acts on the rear axle, that is a common experience; when the brake is applied while the passengers are in the car, it releases itself when they get out of the car. That is experienced to a much greater extent on trucks. It is not exactly a common experience, but it does happen occasionally that a truck releases its brakes when the load is taken off, the brakes having been proprerly locked while the load was on. Another common result is that with the hand-brake applied with the truck empty it becomes impossible to release it after it is loaded. Naturally, a shaft brake, fixed to the frame cannot be affected by spring deflection and hence is the ideal hand brake.

A certain amount of prejudice against the shaft brake originates from the fear that, acting through the drive-shaft, universal-joints, drive gears, differential and axle shafts, it is less reliable and that these parts will be subjected to an abnormal stress from a sudden brake application. Experience shows, however, that failure of rear-axle brakes due to crystallized brake-rods, stuck and rusted pins and burned-out linings is more common than failure of driving parts. The strains to which the driving parts are subjected from shaft brakes, furthermore, are not so severe as is commonly supposed. It can be demonstrated easily that the shock on these parts produced by a sudden application of the clutch at high engine-speed with the gearshift in low or reverse of our modern large-range gearboxes greatly exceeds the brake torque at which the wheels will slide.

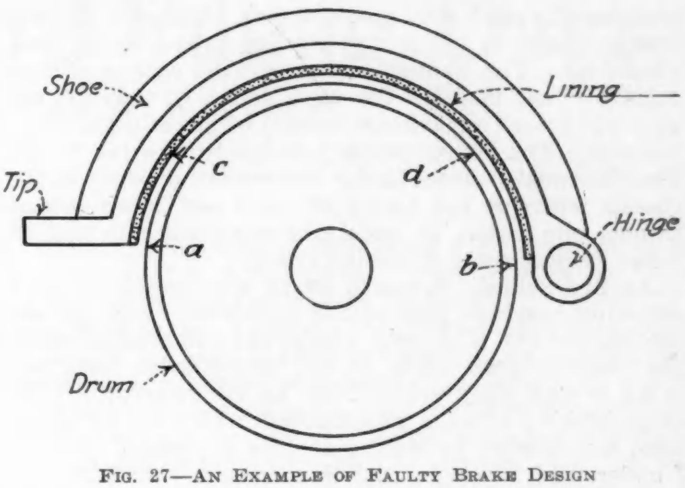


Fig. 27—An Example of Faulty Brake Design

A solid rod is apt to vibrate and crystallize. Cable is one of the first means we used for applying brakes to automobiles, but this has never been entirely satisfactory because cable is apt to fray. The flat-steel strap seems to have a certain amount of promise, except that it is a single strand and, as was found in airplane practice with streamlined solid wire, it is hardly safe. Strap of that sort is brittle and, if it fails, it fails clear across and the rod is broken. Some time ago I had a different style of brake-rod or cable on a small car, that consisted of a form of chain made up of flat brass-links, each link being folded so that the holes in the free ends registered and permitted the next link to be folded through these openings. The experience I had with that substitute was very encouraging. I think a little experimenting in the use of these folded sheet-metal chains will show that they have real possibilities as substitutes for brake-rods. They are extremely flexible, can be made very strong, and offer a very ready means of adjustment.

With regard to the disc brake housed in the differential housing mentioned by Mr. Hastings, granting that it does have a very powerful effect and very long wear and that the brake keeps cool, is not that effect at the expense of the oil? It is not true that the oil in that housing deteriorates very rapidly because it is being burned between the brake surfaces?

Mr. Hastings:—We have operated this brake about 19 months and it was our practice to remove the oil frequently for inspection. We found that it did not deteriorate. During the last part of the run we have had one supply in the housing for 4 or 5 months and we find it in good condition today.

Mr. Horine:—Can you account for that?

Mr. Hastings:—It is because there is a very large area and the heat is distributed over that area.

Mr. Horine:—Will not particles of brake facing, dislodged by friction, be circulated with the oil and do considerable mischief in the gears and bearings?

A. M. Wolf:—It is one of the first fundamentals of brake design to have them absolutely free when they are in the "off" position, and a correct brake should absorb absolutely no power when not in use. Dragging brakes are a prevalent failing and obviously a large factor in fuel-consumption.

To mention a few other means of braking, we can use air when coasting downhill, with the switch off and

working the engine as a compressor. However, the ordinary engine is not a very efficient brake under these conditions. The Saurer engine is built with a sliding camshaft that modifies the valve action so that the engine will absorb a maximum amount of power while turning over. The Saurer people also tried out a fan brake. The fan was mounted under the center portion of the chassis where it had plenty of room and, when it was thrown into action, its resistance was similar to that of a fan dynamometer absorbing energy.

An hydraulically actuated brake was mentioned, and attention has been paid also to hydraulic means of absorbing power. A car was developed in Europe in which the constant-mesh gears of the transmission were encased so that they would form an oil gear-pump. To cause braking effect, a pipe through which the oil circulated was blocked by closing a valve interposed therein. I understand, however, that this was not a success, due to the excessive heating of the oil and the very high pressures encountered; but with modern methods this idea might be revived. Most hydraulic transmissions function as a brake when the control valve is shut or set in a position corresponding with a speed slower than the prevailing one.

The airbrake that uses air as an actuating medium is somewhat old, having been applied on the first Northern four-cylinder car. This car had an air clutch, as well as brakes applied by air-actuated pistons operating the brake-rods. The clutch consisted of a large leather disc forming part of a bellows and was mounted so that, when air was admitted behind it, it would extend slightly forward and come into contact with the rear finished face of the flywheel. The airbrake is now being applied to trailers; it seems that the trailer application will cause both hydraulic and air actuation to become popular. It will be recalled that the Knox-Martin tractor was brought out with an hydraulically actuated brake.

Small reversible high-speed electric motors, acting through a large worm-gear reduction, have been used to actuate the brake-rods. A small button or lever switch-control makes the operation extremely simple; but such a system involves many complications in performing an operation that can be accomplished by very simple means. Unusually large vehicles might be an exception.

With reference to brake adjustments, I mention a device which automatically tightens the brake-rods when their travel is too great. This is done by a ratchet that is held in fixed relation from a cross-member. Movement of the brake-rod beyond a predetermined limit causes the ratchet to rotate a member which is threaded over the brake-rod. This device is borrowed from railroad practice; the slack adjuster, in this case air-actuated, is located on each brake cylinder.

The mounting of the brake cross-shafts deserves consideration, so that they shall be free from binding due to distortion. It was interesting to see how the Renault design obviated any such tendency by its universal mounting. There is one truck on the market that has a cast cross-member which also forms an anchorage for the front end of the rear springs. All the brake cross-shafts are mounted on this member and, due to its unit assembly, there is very little or practically no chance of binding occurring in service.

It is interesting to note the disappearance of the one-time long equalizer bars. They often exceeded the frame width, so that the rods running to the drums would be outside of the frames. Today, the rods are being kept within the frame side-rails. Truck design also is reverting to this method to a large extent. It allows a more substantial brake rigging on the axle.

In the Hotchkiss drive, due to the displacement of the axle under torque, driving and braking stresses, some builders allow for a certain amount of lost motion between the pedal and the final brake-rod. Considering the internal brake, I believe that we should design cams with a small circumferential section or base circle before the shoes are expanded. It naturally would be a more costly cam. This is not necessary, of course, when radius-rods or an anchored torque-tube is used, if the clevis-pin of the brake-lever has the proper location.

Regarding the lubrication of the brake rigging, we see cars today with grease or oil-cups in places that the owner or driver will never bother to reach. In fact, some cannot be reached without crawling under the car on one's back. This refers to brake cross-shafts on the frame, and also to brake shafts on the axle. I am a firm believer in the oilless bushing, of any of the several types, for such locations and I am surprised that all companies do not use them.

# PROGRESS MADE IN GARAGE EQUIPMENT

(*Concluded from page* 66)

in that part of the Country, some trucks and plenty of cars. This shop got all the tractor and all the motor-vehicle business, by virtue of its excellent equipment.

Most dealers will say that garage equipment costs too much. Some of the men in a shop are pretty handy. I have seen some fine engine-stands and other equipment that the mechanics built themselves. If the shop is well tooled-up, one hardly can blame the proprietor if the men make their own equipment. However, the repairmen say they will buy equipment if it is priced reasonably. I saw a portable hoist in a corner of one garage and asked if it ever was used. I was told that one objection to its use is that most cars are fitted with bumpers that prevent this particular hoist from reaching over far enough to lift the engine out.

In regard to the flat-rate system of selling service, suppose I have a shop that is not well tooled-up and some other person has a shop that is very well equipped, and that we both are handling the same make of car. If I use old-time methods in taking a cylinder-head off and removing the valves, and the competing shop has the necessary equipment and can do the job in half the time, it can establish a flat-rate system of selling its service that will appeal far more to the car-owner than my service does. Hence, the equipment plays a direct part in the selling of service by the flat-rate method, and it is one of the essentials that certainly must be considered. I know that the car builders are beginning to take more interest in this. For instance, in laying out the cars they are beginning to plan equipment to go with each operation in service before they settle upon a design. They consider the steps that will be taken in servicing that particular unit. Then they design equipment that will go with that car. That goes to show the amount of thought that is being given to this very important question of garage equipment.

# Malleable-Iron Drilling Data

*By* H. A. Schwartz[1] *and* W. W. Flagle[2]

Cleveland Section Paper

*Illustrated with* Photographs *and* Charts

After commenting upon the two contradictory attitudes toward malleable iron in the automotive industry and outlining its history briefly, the authors discuss the differences between malleable and ordinary gray-iron and supplement this with a description of the heat-treating of malleable castings.

Five factors that influence the machining properties of malleable-iron are stated. These were investigated in tests made with drills having variable characteristics that were governed by six specified general factors. Charts of the results are presented and commented upon in some detail, inclusive of empirical formulas and constants and deductions made therefrom.

The machining properties of malleable-iron is a new subject in engineering literature. C. F. Kettering, past president of the Society, once expressed the belief that the future of engineering would consist of a careful study of all the materials of construction and the selection of the material for a given purpose the properties of which most nearly correspond to the ideal. The department over which I preside was organized with the idea of furnishing the automotive or any interested industry authentic information as to the properties of malleable-iron castings.

Apparently, there are two well-defined and contradictory attitudes toward malleable-iron in the automotive industry, one being decidedly unfavorable. Having had occasion recently to buy a car, I inquired regarding a certain make of machine. The salesman told me that no malleable castings were used in that car. The first thing observable on raising the hood was a malleable casting and, as a matter of fact, the car in question had 31 parts made of malleable-iron. It is unfortunate that anyone should wish to conceal the use of so valuable a material in parts for which it is suited. The opposite viewpoint is held by certain manufacturers. They seem to feel that malleable-iron should give satisfactory service for any purpose whatsoever, and attempt to make everything of it regardless of its suitability.

Fig. 1—Photomicrograph of Gray Iron Magnified 100 Times

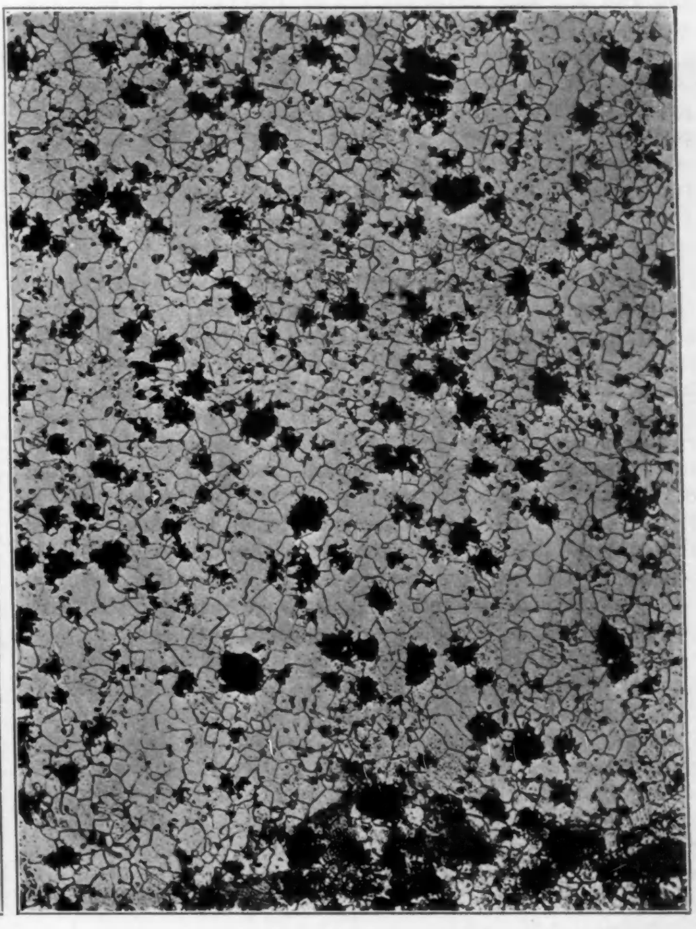

Fig. 2—Photomicrograph of Malleable Cast Iron Magnified 100 Times

The history of malleable-iron dates back to Reaumur, who published a description in 1722 of a system of making it by decarburization. In trying to practice the art of making malleable-iron castings, an American named Boyden discovered a different product in 1826 which is

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now known as black heart malleable. The most recent and accurate figures at my disposal indicate that there are 176 producers of malleable castings in the United States. Others estimate that there are more than 200 producers, but the important ones number between 75 and 80. The principal plants are located in the territory north of the Ohio and east of the Mississippi rivers.

### Malleable and Gray-Iron Differences

Malleable-iron consists of a mass of nearly pure iron or ferrite, through which some 2.0 to 2.5 per cent of carbon is scattered in a spheroidal form in the free state. This particular form of carbon is characteristic of this product only and is known as temper carbon. Gray-iron consists of a metallic mass composed of a mixture of ferrite and pearlite, through which some 3.00 to 3.25 per cent of free carbon is scattered in the form of flaky graphite crystals. It is obvious, as shown by the photomicrographs reproduced in Figs. 1 and 2, that the former conditions produce much less interruption of matrix than the latter. A further consideration is that pure iron is much more ductile than pearlite, which has a corresponding effect upon the two cast products.

The carbon in malleable-iron exists in the geometric form that characterizes it because that carbon is liberated at a temperature when the metal is nearly solid, but the graphite of gray-iron is liberated at a temperature but little below that of the melting point. The fact that it must grow in a nearly solid medium rolls or crushes up the free carbon of malleable-iron into the spheroidal form that characterizes temper carbon. The process of manufacture is first to produce a casting that contains no free carbon and then to heat-treat that casting so as to break up the combined carbon into iron and free carbon.

### The Heat-Treating Process

Two fallacies are encountered frequently with respect to the annealing or heat-treating process. The first is that the process is conducted for the purpose of eliminating carbon and that, therefore, the surface of the metal must differ widely in properties from those of the center. The elimination of carbon from the surface metal is a mere incident in the process and affects the metal but slightly, increasing the ductility of the product a little. The primary purpose of the heat-treatment is the separation of cementite into ferrite and carbon, and this process does not proceed more rapidly or more completely at the surface than within. Malleable castings, when machined, therefore possess properties that are comparable with those of unmachined castings.

The second fallacy is that the annealing reaction is similar to that used for the annealing of steel and, therefore, the malleable-iron manufacturer is taking too much time for this process. An automotive engineer of my acquaintance once insisted most strongly that we were wrong in taking 9 days to heat-treat castings. He said he would prove this to us by taking a casting in the evening and returning it completely annealed in the morning. That was more than a year ago and he has not yet returned. Steel can be annealed in a few hours, but the various stages in the graphitizing heat-treatment require definite and specific times and cannot be executed in a shorter time interval. All malleable-iron producers would arrange to graphitize the carbon completely overnight if that were possible. The process constantly is subject to study and experiment with a view to decreasing the time involved. So far, however, no great reduction has been found possible.

An attempt to hurry the annealing process results in the user obtaining an inferior product which impairs the reputation of the producer. Those who purchase material should remember that long annealing processes are executed at the expense of the manufacturer; obviously, they would not be carried out if they were not essential to the satisfactory completion of the product. It is absolutely necessary that demands for malleable-iron products be adequately anticipated to allow sufficient time for this process of manufacture.

The physical properties of normal malleable-iron under various circumstances are covered in my previous papers entitled Malleable Iron as a Material for Engineering Construction[3]; Some Physical Constants of American Malleable-Iron[4]; and The Effect of Machining and of Cross-Section on the Tensile Properties of Malleable-Iron.[5]

### Machining Properties

The machining properties of malleable-iron are the subject of experiment in the research laboratory of the National Malleable Castings Co. The behavior of twist drills on other products has been the subject of experiment more especially by B. W. Benedict and W. P. Lukens, who gave data obtained in drilling gray-iron in their report entitled An Investigation of Twist Drills.[6] The only data on malleable-iron of which we have knowledge were secured in a very short investigation by Edwin K. Smith and William Barr, and reported in a paper on The Relation Between Machining Qualities of Malleable Castings and Physical Tests.[7]

Feeling that further work was requisite on drilling stresses when cutting malleable-iron, it was decided to begin the study of machinability in general by an investigation of these stresses. The variables, the effects of which are to be studied, include the following, those numbered from 1 to 6 being in reference to the drill and those from 7 to 11 in regard to the properties of the material.

(1) Diameter
(2) Rate of feed
(3) Speer
(4) Point angle
(5) Clearance angle
(6) Helix angle
(7) Chemical composition
(8) Tensile-strength
(9) Elongation
(10) Brinell hardness number
(11) Shore number

Since the life of the drill is not under observation, the chemical and physical properties of the drill steel were not significant and were assumed to be constant throughout. The effect of cutting compounds was not within the scope of the investigation.

The experimental procedure was to determine on the Olsen universal efficiency machine the torque and thrust of drills operating under various predetermined conditions. Reference is made to a paper by T. Y. Olsen on An Efficiency Testing-Machine for Testing Taps and Dies.[8] All the drills used were of high-speed steel, made and ground by the Cleveland Twist Drill Co. No drill

[3] See *Transactions of the American Foundrymen's Association*, vol. 27, p. 373.

[4] See *Proceedings of the American Society for Testing Materials*, vol. 19, part 2, p. 247.

[5] See *Proceedings of the American Society for Testing Materials*, vol. 20, part 2, p. 70.

[6] See Bulletin No. 103, Engineering Experiment Station of the University of Illinois, 1917-1918, vol. 15, No. 13.

[7] See *Transactions of the American Foundrymen's Association*, vol. 28, p. 330.

[8] See *Proceedings of the American Society for Testing Materials*, vol. 14, part 2, p. 541.

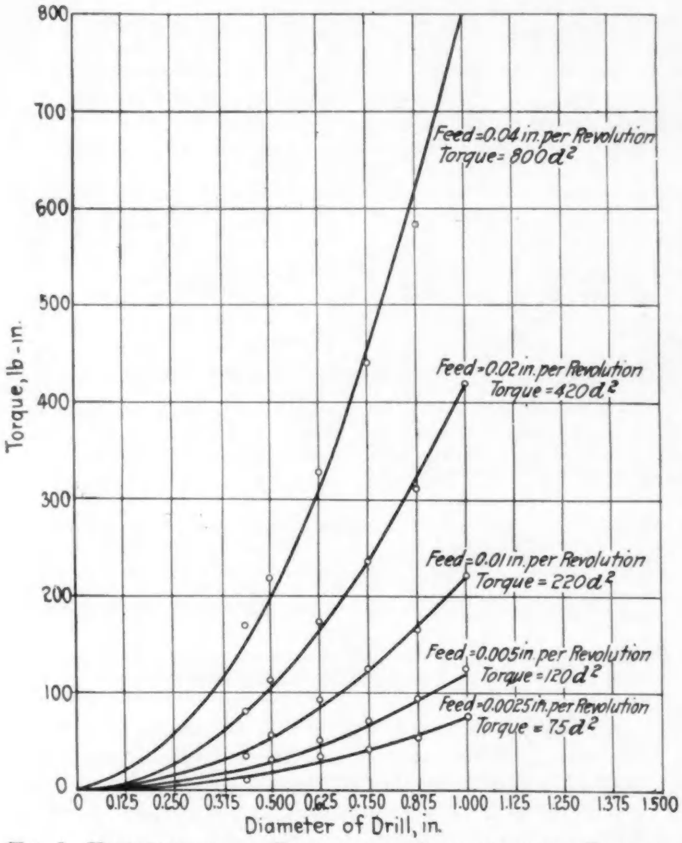



FIG. 3—VARIATION IN THE TORQUE WITH CHANGES IN THE DIAMETER OF THE DRILL AND THE RATE OF FEED

was used to a point where a change in stress that could be detected was produced. This factor was checked by drilling a standard malleable-iron piece from time to time, under standard conditions, as the investigation progressed.

For the investigation of items Nos. 1, 2 and 3, a large amount of malleable-iron from a single heat and a single annealing pot was available. The investigation comprised the drilling of this material with twist drills of standard form having diameters of 7/16, ½, ⅝, ¾, ⅞ and 1 in. at feeds of 0.0025, 0.0050, 0.0100, 0.0200 and 0.0400 in. per revolution at a speed of 240 r.p.m.; and the drilling of the same material with a standard ¾-in. twist-drill at the same range of feeds at speeds that were, as nearly as practicable, 40, 80, 160, 320 and 640 r.p.m. A few runs were made with drills of other diameters to corroborate the conclusion that the effect of speed did not vary greatly with drill diameter.

For the investigation of items Nos. 4, 5 and 6, a new lot of malleable-iron was employed which, from tests made with standard twist-drills, was known to be identical with the first in resistance to drilling. Instead of using the standard drill, which has a point angle of 118 deg., a clearance angle of 12 deg. and a helix angle of 27.5 deg., nine special drills, each ¾ in. in diameter, were provided, that had point angles of 98, 118 and 138 deg. and clearance angles of 5, 10 and 15 deg., respectively. The helix angle was 27.5 deg. in each case. A straight-fluted ¾-in. drill having standard point and clearance angles was provided also.

For the investigation of items Nos. 6 to 11 inclusive, 179 specimens of regular and special malleable-iron were available. These were made by the several plants of the National Malleable Castings and the Eastern Malleable Iron companies and by the Dayton, the Northern, the Erie and the Trenton Malleable Iron companies. Our thanks are due these organizations for placing at our disposal material of divers origins and processes of manufacture. This material represented practically the complete range of quality commercially attainable in the product. Some low-grade specially made material was also used. The metal was all completely graphitized. The chemical composition and tensile properties were determined by the foundry. The Brinell hardness and Shore numbers were obtained in our own laboratory.

Each material was tested with two standard ½-in. drills at 240 r.p.m. and a 0.005-in. feed. The material was drilled also by each of two ½-in. drills running at 240 r.p.m. with a constant pressure of 220 lb. on the drill point; the torque and the penetration per revolution were recorded. It is obvious that the detailed results of such an investigation are too voluminous to be given and the essential data have therefore been reduced to graphic form.

### GRAPHIC TEST-RESULTS

Fig. 3 shows the relation of torque to diameter and feed as a series of parabolas, one for each rate of feed, correlating the diameter and the torque.

Fig. 4 shows the relation of the thrust to the diameter and the feed as a series of flat curves, one for each rate of feed, correlating diameter and thrust. The thrusts developed by the smaller drills, especially at low rates of feed, are so low in value as to render the data somewhat uncertain.

Fig. 5 shows the relation between the torque and the thrust of a ¾-in. drill as related to the speed, a separate line being shown for each rate of feed. A few tests for ½ and 1-in. drills were conducted, tending to correlate

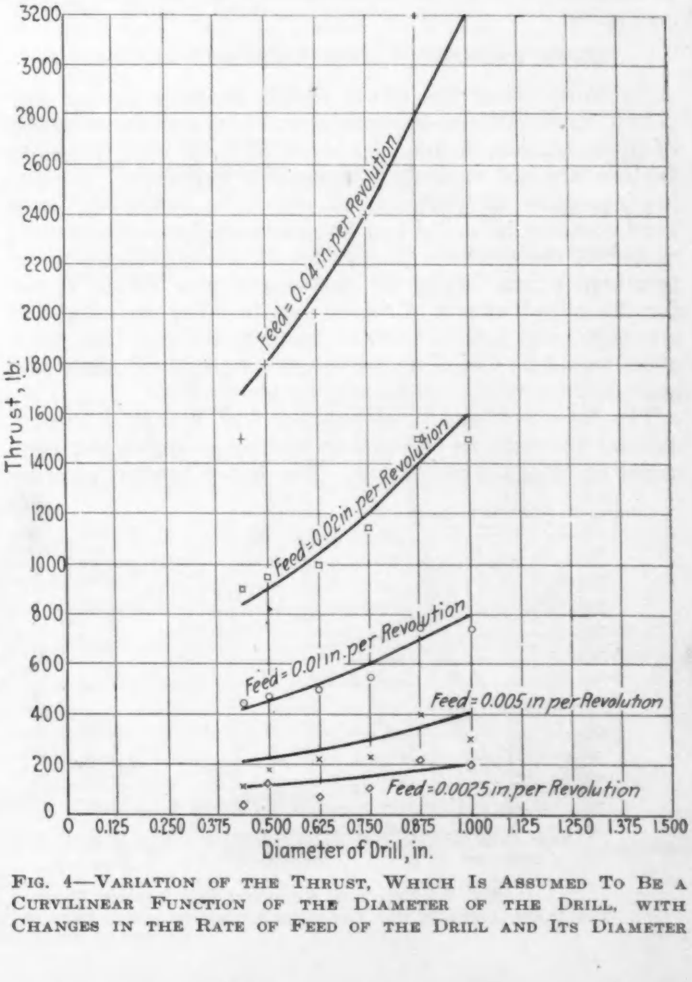



FIG. 4—VARIATION OF THE THRUST, WHICH IS ASSUMED TO BE A CURVILINEAR FUNCTION OF THE DIAMETER OF THE DRILL, WITH CHANGES IN THE RATE OF FEED OF THE DRILL AND ITS DIAMETER

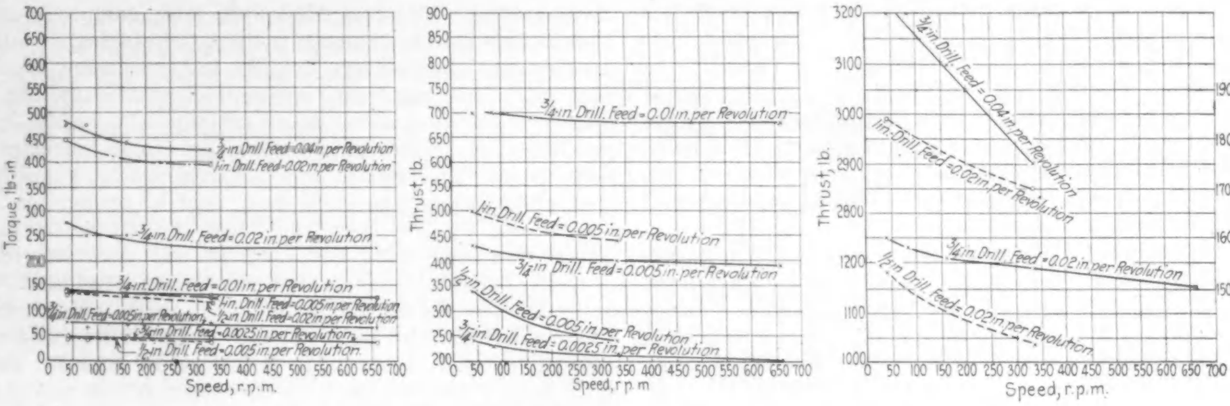



FIG. 5—TORQUE AND THRUST OF A 3/4-IN. DRILL AT DIFFERENT SPEEDS AND FEEDS
Results of Some Tests Made with 1/2 and 1-In. Drills That Were Conducted To Correlate and Corroberate the Conclusions Based on the Tests of the 3/4-In. Drill Are Also Presented

and corroborate the conclusions based upon the 3/4-in. drills.

Fig. 6 shows the torque and the thrust of drills of various point and clearance angles as related to speed. Each chart applies to a drill of a given point-angle, the thrust or the torque being plotted against the speed. A different symbol was used for each of three clearance-angles.

Fig. 7 shows the torque and the thrust of the straight-fluted drill plotted against the speed at two different rates of feed and, for comparative purposes, the same data for a standard twist drill. Thus, Figs. 3 to 7 record the available data on the drilling conditions as a variable and permit fairly accurate conclusions as to the load conditions on drills of known diameter and form, working on the standard material under known conditions of feed and speed.

## EFFECT OF THE MATERIAL BEING MACHINED

In considering the effect of the material being machined as comprised in items Nos. 7 to 11 of the program of investigation, it must be borne in mind that these five factors are not entirely independent variables. Assuming complete graphitization, which is justifiable with good commercial metal and known to apply to our present material, the physical properties of the product are determined primarily by its carbon-content. Were it not for the minor effects of variations in other chemical elements present and in thermal history, the last four variables would be functions of the carbon-content alone and bear definite relationships among themselves.

The tensile-strength, elongation and Brinell hardness number increase as the carbon decreases regularly, even under commercial conditions. The Shore number is prac-

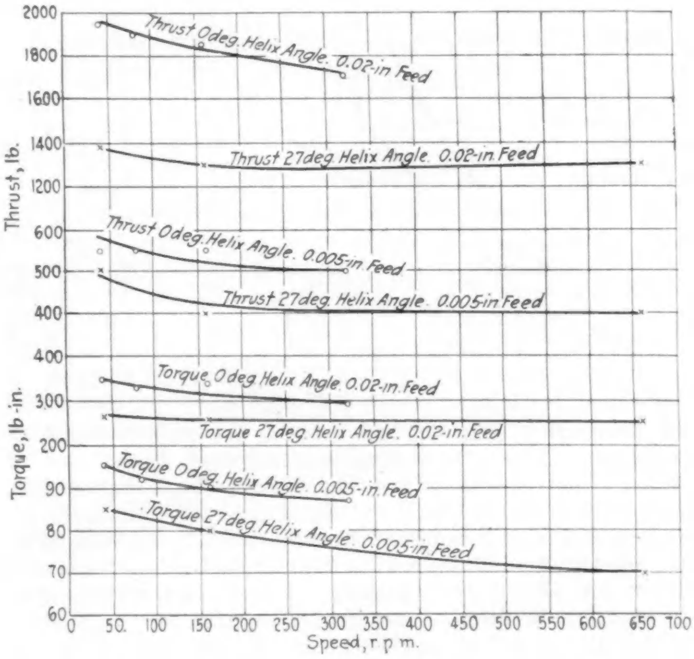



FIG. 7—TORQUE AND THRUST OF STANDARD TWIST AND STRAIGHT FLUTED DRILLS IN MALLEABLE IRON

tically constant for all samples investigated. Therefore, it was to be expected that graphs plotting drilling stresses against items Nos. 7 to 10 would be geometrically similar, and the Shore number would find no application in determining the physical properties of the material. Our tests were all made under such conditions that only material at least 1/8 in. from the surface was examined. Under

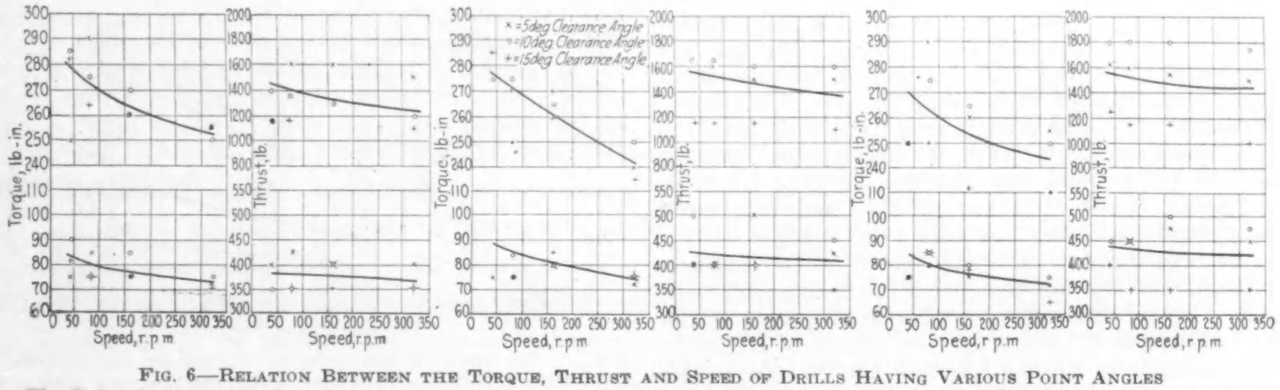



FIG. 6—RELATION BETWEEN THE TORQUE, THRUST AND SPEED OF DRILLS HAVING VARIOUS POINT ANGLES
The Point Angle in the Two Sets of Curves at the Left Was 98 Deg., for the Middle Pair It Was 118 Deg., and for the Curves at the Right It Was 138 Deg.

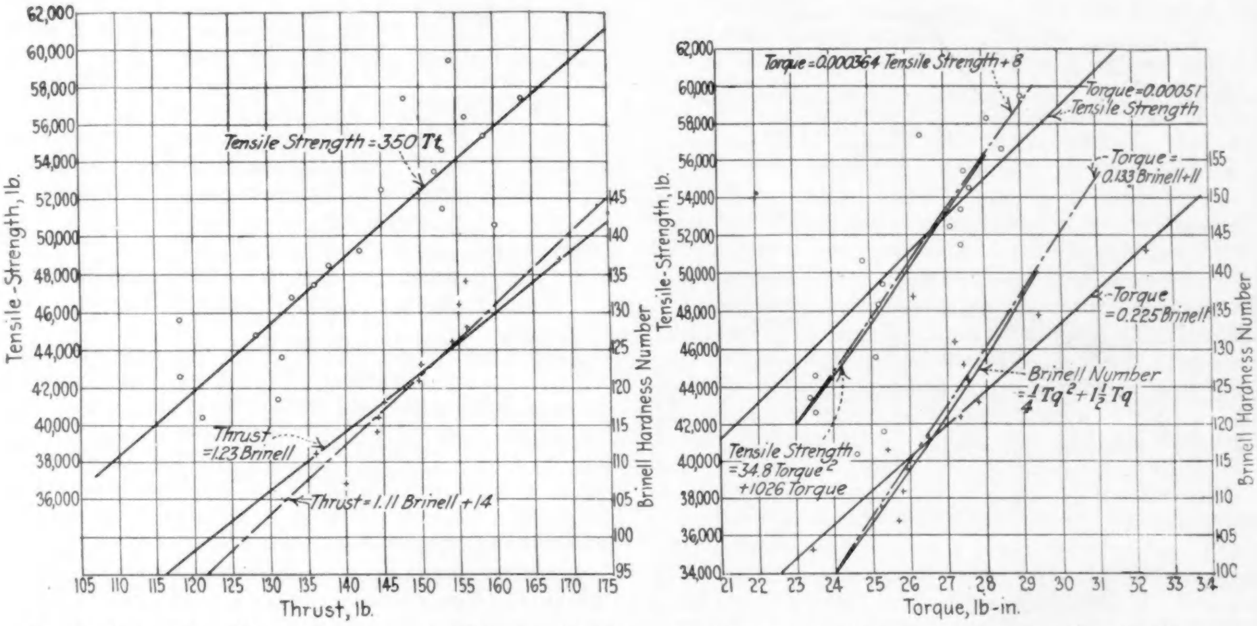



FIG. 8—AT THE LEFT CURVES SHOWING THE RELATION BETWEEN THE THRUST OF A ½-IN. DRILL AND THE BRINELL HARDNESS NUMBER AND THE TENSILE-STRENGTH OF THE IRON AND AT THE RIGHT SIMILAR CURVES FOR THE TORQUE

these circumstances, no connection between decarburization and machinability could be traced, even if it existed. Item No. 7 can thus be dismissed with the statement that items Nos. 8 to 10 are expressions of this variable. Item No. 11 can be discarded on account of the following considerations affecting the Shore number. The usual physical properties of malleable iron are determined primarily by the relative amount of ferrite and temper carbon in the mass. The Shore test, however, is made upon an almost microscopic area comprising only ferrite, and it is therefore a measure of the properties of the ferrite only. These properties would be affected but little by the presence of the usual amounts of alloy.

In Fig. 8 the thrust and the torque of the ½-in. drill running at 240 r.p.m. are plotted against the tensile-strength and Brinell hardness number, each point representing the average drill-stress for a group of specimens of constant strength or hardness as the case may be. A similar compilation against elongation is omitted as superfluous, since elongation and tensile-strength are interdependent and the latter property is measurable more accurately. Omitting the derivation of the several formulas in the interest of brevity, the data of Figs. 3 to 8 lead us to the following general conclusions that apply to completely annealed malleable-iron castings.

Let

$a$ = A constant depending upon the feed
$B$ = The Brinell hardness number of the malleable iron
$b$ = A constant depending upon the diameter of the drill
$d$ = Drill diameter in inches
$f$ = Rate of feed in inches per revolution
$P$ = The thrust in pounds
$s$ = Speed in revolutions per minute
$T$ = The drill torque in pound-inches
$U$ = The ultimate-strength of the malleable iron in pounds per square inch
$W$ = The work done in drilling, in foot-pounds per cubic inch

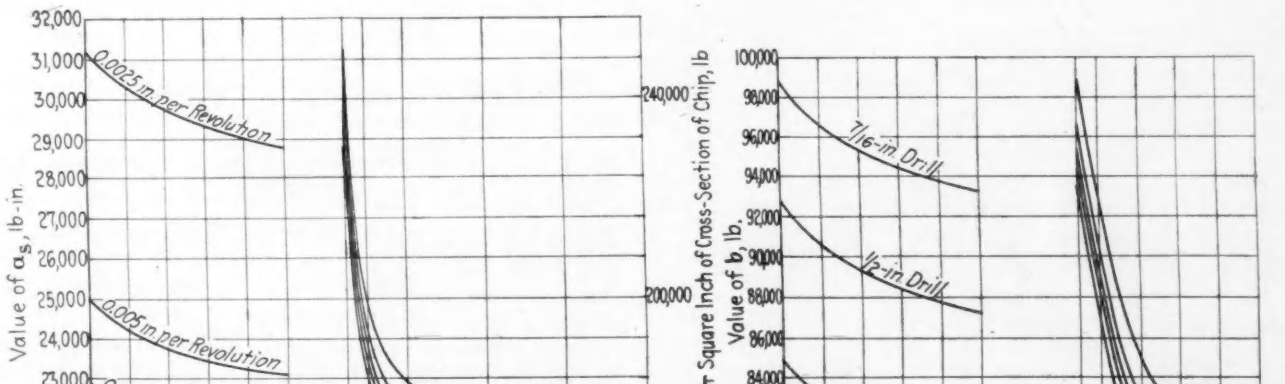



FIG. 9—CURVES GIVING THE VALUES OF $a_s$, $k_s$ AND $b$ FOR VARIOUS SPEEDS

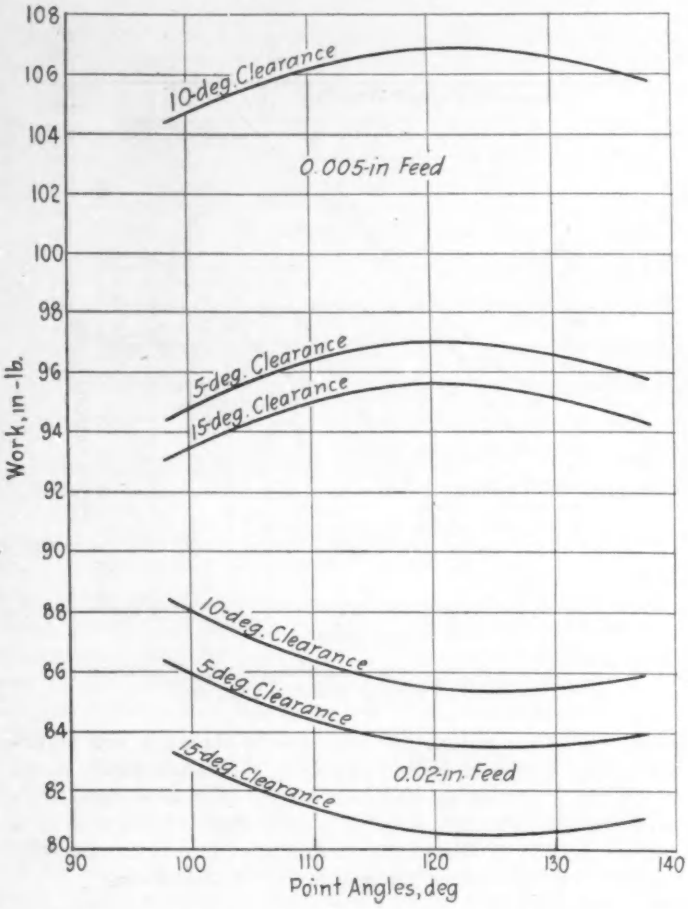



FIG. 10—CURVES GIVING THE AMOUNT OF WORK REQUIRED TO PENETRATE MALLEABLE IRON A DISTANCE OF 1 IN. WITH ¾-IN. HIGH-SPEED DRILLS OF DIFFERENT POINT AND CLEARANCE ANGLES

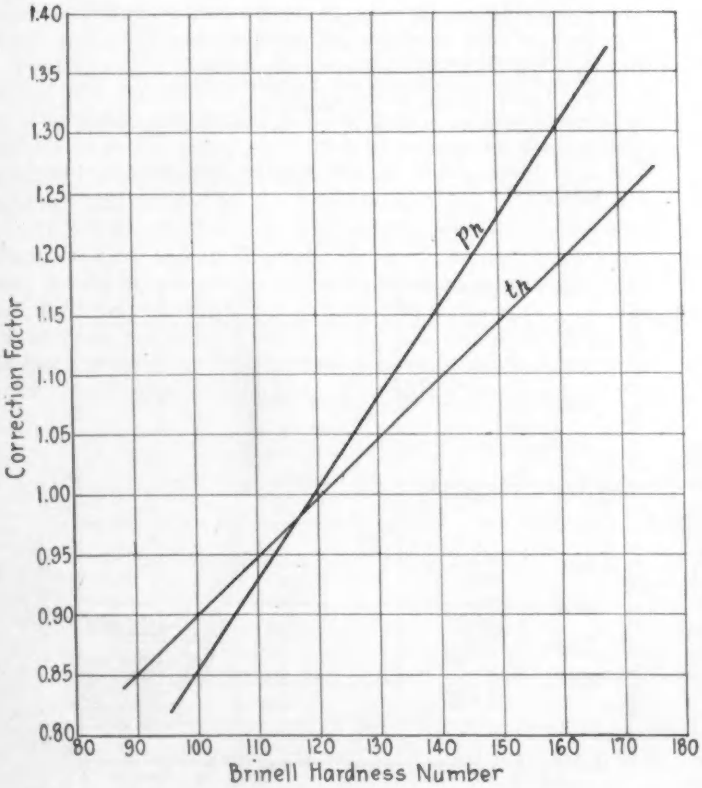



FIG. 11—CORRECTION FACTORS FOR $p_h$ AND $t_h$ FOR VARIOUS BRINELL HARDNESS NUMBERS

The values of $a$ and $b$ are shown in Fig. 9. Then, for drills of standard form, we have

$$T = (0.0049\,B + 0.409) \times (167.06/s^{0.04597})\ a \times fd^2$$
$$P = (0.00755\,B + 0.0952) \times (937/s^{0.03450})\ b \times fd$$
$$W = 8\ (0.0049\,B + 0.409) \times (167.06/s^{0.04597})\ a + (0.00755\,B + 0.0952) \times (937/s^{0.03450}) + (4b/\pi d)$$

where $T$, $P$ and $W$ are in terms of Brinell hardness number. Or, in terms of ultimate-strength, we have

$$T = (0.0000135U + 0.297) \times (167.06/s^{0.04597})\ a \times fds^2$$
$$P = (U/52{,}000) \times (937/s^{0.03450})\ b \times fd$$
$$W = 8\ (0.0000135U + 0.297) \times (167.06/s^{0.04597})\ a + [(U/52{,}000) \times (937/s^{0.03450})] + (4b/\pi d)$$

To simplify these equations, let

$$(167.06/s^{0.04597})\ a = a_s$$
$$(937/s^{0.03450})\ b = b_s$$
$$0.0049B + 0.409 = t_h$$
$$0.00755B + 0.0952 = p_h$$
$$0.0000135U + 0.297 = t_u$$
$$U/52{,}000 = p_u$$

Substituting these values, we have

| In Terms of Hardness | | In Terms of Ultimate-Strength |
|---|---|---|
| $T = t_h \cdot a_s \cdot fd^2$ | = | $t_u \cdot a_s \cdot fd^2$ |
| $P = p_h \cdot b_s \cdot fd$ | = | $p_u \cdot b_s \cdot fd^2$ |
| $W = 8t_h\, a_s + 4\ (p_h \cdot b_s/\pi d)$ | = | $8t_u\, a_s + 4\ (p_u b_s/\pi d)$ |

For convenience, values of $a_s$ are plotted at the left of Fig. 9; of $b_s$ at the right of the same illustration; of $t_h$ and $p_h$ in Fig. 10; and of $t_u$ and $p_u$ in Fig. 11. From these values, $T$, $P$ and $W$ can be calculated for known values of the several constants.

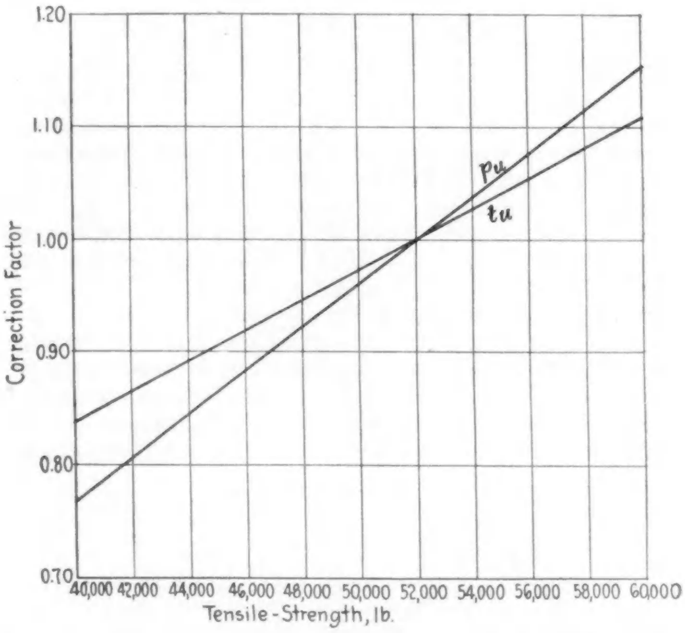



FIG. 12—CORRECTION FACTORS FOR $p_u$ AND $t_u$ FOR DIFFERENT TENSILE-STRENGTHS

That the torque should be proportional to the feed and the square of the diameter; and that thrust is proportional to feed and diameter; and the derivation of $W$ from $T$ and $P$, are based on considerations of applied mechanics. The numerical constants are, of course, purely empirical.

In Fig. 12 the relation between work and form of drill point is summarized, as calculated from Fig. 6. Generalizations as to drill form are as yet hardly warranted. From considerations of mechanical efficiency, an increase

in the helix and the clearance angles to the highest values consistent with drill wear and strength would seem to be indicated. The effect of the point angle seems unexpectedly small.

By substituting in the formulas for $t_h$, $p_h$, $t_u$ and $p_u$, the extreme values of $B$ and $U$ likely to be encountered it will be observed that the difference in torque between the weakest and strongest irons reasonably likely to be obtained is approximately 30 per cent of the average value. The thrust may vary perhaps 40 per cent between the strongest and weakest commercial irons.

## THE DISCUSSION

W. R. STRICKLAND:—What progress has been made in casting thin sections?

H. A. SCHWARTZ:—Malleable iron originally was developed for use in small, thin castings. The recent tendency has been toward heavier sections. I have seen castings where perhaps 20 would be required to weigh 1 oz.; however, such work is very unusual. To avoid cooling strains that may crack the casting, it is necessary to avoid abrupt changes from thin to thick sections.

A MEMBER:—The aluminum foundries have become expert in the use of chills to prevent the cracking of castings. Is there any such art in the making of malleable castings?

MR. SCHWARTZ:—Chills have been used for many years to keep out cracks and also to remove large shrinks. Perhaps I should have spoken of this more fully. One can make perfectly good malleable-iron and still make a very bad casting. For instance, in the automotive trade malleable-iron, when properly produced, is the best possible material for making hubs; but when improperly made, it is about the worst. The reason is that if the work is entrusted to a producer who desires to make the most castings for the least money irrespective of quality, a hub may be produced that is 95 per cent hub and 5 per cent air; the 5 per cent represents the volume of shrink. Being at the point of junction of the barrel of the hub with the flange, these shrinks weaken it at its most critical point and easily may be a cause of failure. The remedy is not in better iron, but in better foundry practice in avoiding shrinkage by the use of suitable feeders. This method, however, costs money and is not likely to be practised by those who feel it necessary on account of competition to cut prices to the limit.

A MEMBER:—Have you experimented with malleable-iron pistons? How much will electric-furnace practice increase the cost of castings? Is the element of manganese considered important?

MR. SCHWARTZ:—Our company has not experimented with pistons. I think malleable-iron possesses no particular advantage. I will not undertake any statement as to the relative cost of electric-furnace and air-furnace malleable. Electric-furnace malleable has been sold continuously in competition with the good grades of air-furnace metal. Manganese is of importance on account of the unavoidable presence of sulphur; but I warn any consumer against specifying the chemical properties of the material he desires to buy. The user buys the material on account of its physical or engineering qualities and is interested in the material possessing the qualities that are required. He is usually not well informed as to the method of manufacture, certainly not as well informed as the producer, and should refrain from telling the latter how to secure the results.

A MEMBER:—Is malleable-iron suitable as a bearing metal, for heavy loads?

MR. SCHWARTZ:—Malleable-iron is not suitable for bearings, but there is a specific case in the design of the trucks under a freight or passenger car where the journal-boxes ride against the truck column. In this construction malleable-iron is used. In other words, its resistance to wear is good enough so that its utility for purposes where wear resistance is an incident will not be destroyed. It should not be recommended purely as a bearing material, so far as we know now.

FERDINAND JEHLE:—A piston must be made of good bearing metal, and have walls of such cross-section that they will conduct away some of the heat. The mere fact that it might be possible to make it very thin would not mean that it would be a good piston.

MR. SCHWARTZ:—I do not see, offhand, how one could cast a thinner piston of malleable-iron than of gray-iron, but I think one would have no trouble with it.

E. T. BIRDSALL:—Does it take longer to anneal castings weighing 50 to 60 lb. and from ½ to 1 in. thick, than thinner sections? Would it cost more or less to machine malleable than steel castings?

MR. SCHWARTZ:—The heaviest casting of which I have knowledge weighed 896 lb.; it was a transmission housing on a military tractor. A thickness of ½ in. is not considered thick; about 1 in. is considered moderately thick, although much heavier sections have been made. Thickness, as such, has no connection with annealing time, although for other reasons fairly heavy castings take longer to anneal than small ones. I think that, as a general rule, a malleable casting of ordinary size, including machining, will cost less than the corresponding steel casting, the difference being largely in the cost of the machining.

A MEMBER:—You mentioned the wear of malleable axle-housings on railroad-car service. How does the wear compare with that of steel journal-box housings? Is cast-iron used for this purpose?

MR. SCHWARTZ:—Cast iron has been used, although malleable-iron makes the better journal-box. Cast-steel journal-boxes have been made by our company, but they are not common. I know of no figures showing the comparative wear of the several materials. We are now working along these lines.

A MEMBER:—Is there any way of filling porous places in malleable castings?

MR. SCHWARTZ:—They can be filled with compounds such as Smooth-On. They can be filled also by bronze welding or brazing with the acetylene torch. The producer can fill the pores by acetylene welding with white cast-iron and reannealing the castings. In view of the difficulty of producing perfect welds, it is doubtful whether this practice should be resorted to at critical points in a casting. Our company is opposed to it.

A MEMBER:—Does your company request manufacturers to state where they wish the sprue put?

MR. SCHWARTZ:—Not in exactly that form. As a rule, the buyer of a casting is not sufficiently well acquainted with foundry practice to have an opinion of value on this point, but we welcome an expression from him as to where the important points are from which it is necessary to exclude shrinkage.

A MEMBER:—Is it possible to nickel-plate malleable castings?

MR. SCHWARTZ:—That is a very common practice.

# Research Topics and Suggestions

*THE Research Department plans to present under this heading each month a topic that is pertinent to the general field of automotive research, and is either of special interest to some group of the Society membership or related to some particularly urgent problem of the industry. Since the object of the department is to act as a clearing-house for research information, we shall be pleased to receive the comments of members regarding the topics so presented, and their suggestions as to what might be of interest in this connection.*

## GEARS

THE subject of gears and gear performance is too broad to permit of more than a few suggestions as to some of the phases that appear to need further study from the research standpoint, particularly in view of the demand for a certain amount of standardization in gear practice in the automotive field.

For the purpose of this discussion, we shall omit all of the many questions of gear design, such as tooth contour, pitch and pressure angle, all of which have been dealt with at length in the literature and probably are well known to all who have made a study of the design and production of high-duty gears.

The topics that seem to have received the least attention in the literature of the industry, and probably have received correspondingly less attention from engineers, are those of gear performance. The performance requirements of gears for automotive use may be classed under the headings

(1) Strength (3) Quietness
(2) Wearing quality (4) Mechanical efficiency

The relative importance of these four qualities depends entirely upon the type of gear and class of service. For instance, in tractor gearing, wearing quality is likely to be of the greatest importance, whereas for a passenger-car transmission, adequate strength and reasonable quietness may be demanded, and for front-end gears, quietness alone may be the major requisite.

A review of the published material on the foregoing topics reveals a disappointingly small amount of important matter that deals directly with them.

### Strength

Nearly all of the many formulas for computing the strength of gears are based on the formula published by Wilfred Lewis in a paper that appeared in the *American Machinist*, May 4, 1893. The author states that at that time there were some 48 different formulas for the strength of gear teeth, differing by factors of as much as 500 per cent.

The Lewis formula, as commonly used, consists of two parts, one of which applies to static loads, the other to the increase in tooth pressure with speed. The former part is an exact, or "rational" formula based on certain assumptions. If a given load is applied uniformly across the tip of a tooth, in a tangential direction, the maximum fiber stress is given by the formula with only the assumptions necessary in computing the stresses in a beam. However, these assumptions are open to question, and the formula cannot be more nearly correct than the assumptions on which it is based.

The second portion of the formula is a factor depending upon speed and is based on the assumption that the maximum tooth-pressure for a given average torque increases in proportion to the speed with an arbitrary factor of proportionality, the basis of which the author does not give. This portion of the formula is not exact or rational, but is supposedly based on the results of experiments that must have been made prior to 1893; probably, therefore, on gears of entirely different types from present-day automobile gears. It is hardly to be expected that the Lewis formula, including this factor, would apply to modern practice.

So far as we have been able to determine from published data, there are no formulas for gear strength in use at present that give entirely trustworthy results, and most of the gears designed primarily for strength seem to be laid out on the basis of experience alone. This is by no means a satisfactory condition, and it cannot be remedied until we can provide answers to the following questions:

(1) Are the assumptions on which the Lewis formula is founded well based, and what modifications are necessary to make it, or any similar formula that may be developed, more readily applicable to modern practice?

(2) What is the effect of speed in increasing the pressures on gear teeth beyond those due to the average torque? Is this increase proportional to the speed or to the square of the speed? To what extent does it depend upon changes in angular velocity, and on the moment of inertia of the rotating parts?

It seems rather surprising that so little experimental work has been done on the latter problem. While a few isolated experimenters have improvised means for measuring the relative and actual angular velocities of a set of gears, so far as we have been able to learn, no instruments or methods have been developed which would facilitate a general study of this subject Yet without such a study it is impossible to determine the effect of speed on the maximum stresses set up in gearing A few of the more recent articles dealing with this phase of the subject are

The Strength of Gear Teeth. Guido H. Marx, *Journal of the American Society of Mechanical Engineers*, Vol. 35, p. 109.

This article describes an elaborate series of tests of strength of cast-iron gears under load at various speeds. Results are tabulated at length, and causes given for the relation of load to breaking stress.

A peculiar result is that breaking stress is a minimum at a moderate speed, and increases with higher speeds on the two curves given. Comparing the actual figures of breaking stress for speeds of 0, 100, 300 and 600 ft. per min. with the Lewis formula, actual factors of safety run from 5.5 to 12.2. Static breaking-stresses were measured for one-tooth and for two-tooth engagement by loading the cast-iron gear against a steel gear with teeth cut away except for the number required for contact. Breaking stresses were found to be very much in excess of those to be assumed from the Lewis formula. The author criticises the Lewis formula, showing several important omissions, such as neglect of distribution of stress between different teeth and neglect of angularity of load application. This is an important and rather carefully prepared article, and should be of interest in connection with any study of gear strength.

Tests of Strength for Gear Teeth. Andrew C. Gleason, *Machinery*, January 1914, p. 382.

Mr. Gleason describes the results of a special test for the breaking strength of gear teeth, the gears being held in a special chuck and one tooth subjected to load. A special micrometer was used to detect distortion. Figures are given for a gear, 1-in. face, 2⅓-in. pitch diameter with 14 teeth of 6 pitch, loaded at the tip of the tooth. The breaking stress varies on six different materials with different heat-treatment from 9000 to 22,450 lb. per sq. in.

Strength of Gear Teeth. S. J. Berard, *American Machinist*, Nov. 27, 1919, p. 925.

This is a discussion of gear-tooth strength on the basis of the Lewis formula. The author points out that the strength

is not the same for gears of different sizes. He gives a chart of values of $T$ and $F$ in the Lewis formula, where $T$ is the beam thickness and $F$ is the face width. He uses Barth's formula for the effects of speed.

Approximate Method for Determining the Strength of Gear Teeth. Willard A. Thomas, *American Machinist*, Aug. 7, 1919, p. 273.

The author quotes the Lewis formula, and develops a more simple practical formula making some simplifying assumptions. The proposed new formula might be of value from its simplicity when applied to cast-iron or bronze gears used in general machinery practice. It is not capable of general application.

Convenient Forms of the Lewis Formula for the Strength of Gear Teeth. J. H. Carver, *Machinery*, October, 1914, p. 99.

This is a brief paper giving some examples and supposed simplifications of the Lewis formula.

### Wearing Quality

For classes of gearing operating at anywhere near full load, for long periods, the rate of wear, rather than strength, may be the determining factor in design. It is, perhaps, natural that not much experimental work has been done on the subject of gear wear, since it is affected by important circumstances over which the designer has little control, such as the kind of lubricant and the amount of abrasive material present, as well as the sort of service required. Some valuable contributions on this subject have been made by Joseph Jandasek, in a series of articles entitled Gear-teeth Sizes from the Standpoint of Durability, published in *Automotive Industries* June 10 and 17, 1920, pp. 1305 and 1402.

In these papers the author points out some of the limitations in the computation of gear dimension on a basis of strength, as for instance the effect of tooth deflection on stress distribution, and maintains that unit surface-pressure is of equal importance with total pressure in determining rate of wear. Wear is not proportional to unit pressure, but is small up to a certain critical pressure beyond which it increases very rapidly.

The author calls attention to the various types of contact between curved surfaces that occur in machinery practice, such as plain and roller cams, ball and roller bearings and gear teeth, and taking as his basis the formulas of Hertz, develops an expression for the compressive stress at the contact surface for the several forms of contact.

He discusses at some length the relations between the results of the Lewis formula and practice with gears of different materials, and proposes a speed increment proportional to the square of the speed instead of to the first power as proposed by Lewis, and states his reasons for considering this a more rational assumption.

A discussion of the comparative merits of case-hardened versus tempered gears follows. Case-hardened gears are recommended for constant-mesh service, and tempered gears for change-speed. While case-hardened gears are less subject to pitting than tempered gears, pitting can be reduced in tempered gears by reducing the unit pressure and using a harder temper. The author develops formulas for unit pressure on gear teeth, showing that this pressure depends upon the elasticity of the materials. Gray-iron and bronze are cited as good gear materials for moderate loads owing to their high deformability.

An example is given to illustrate the calculation of maximum compressive stress in a set of gears.

A second series of papers on gearing, Gearing Calculations by the Compressive Stress Method, by Joseph Jandasek, published in *Automotive Industries* Sept. 15 and 22, 1921, pp. 512 and 564, is a further development of the subject of the relation between unit pressure and rate of wear. Resistance of metals to wear depends upon (*a*) hardness where the particles of metal are not readily displaced and (*b*) toughness where the particles of metal are not easily removed, even if displaced.

The remainder of the paper deals with one of the above-mentioned topics, the effect of speed on the strength of gears. The author discusses at some length the causes of increase in the load with an increase in the speed and proposes a new empirical formula based on his own observations of variation in angular velocity. Increments in load are due to

(1) Accelerations, both positive and negative, due to
    (*a*) Irregularity of tooth outline
    (*b*) Variable gear-tooth and surface deflection
    (*c*) Wear
(2) Accelerations, or deflections due to shock or oscillation (Resonance effects)

The author prefers his formula, in which the load is assumed to increase with the square of the speed, to that of Lewis, in which the load is assumed to increase as the speed, because

(1) Accelerations must increase *with higher power of speed*, about the square; moreover, no gears are safe at high speeds
(2) Weight of parts increases with load
(3) Deflection increases with load

He then develops a new formula for load capacity differing from one given in a previous paper, and based on the assumption of a maximum allowable comprehensive stress at the tooth face, and discusses the influence of

(1) Gear ratio
(2) Quality of material
(3) Application to helical gears

These papers are too long and too important to one interested in the subject of gear wear, to permit an adequate review here. They should be read with care.

Another paper dealing with this subject is An Investigation of Tooth Wear with Automobile Gear Steels, by E. R. Ross, in *Automotive Industries*, Nov. 3, 1921, p. 865, in which the author reports the results of tests of gears of three classes, tempered gears with two degrees of hardness, and case-hardened gears still harder than the former. He recommends a scleroscope hardness of 75 or over to secure minimum wear.

An article on Spur-Gear Erosion, by F. W. Lanchester, in *Engineering* for June 17, 1921, provides an interesting discussion of the causes of spur-gear erosion, and suggests remedies.

The author points out that rolling in place of sliding contact between spur gears has been considered the ideal condition. Contrary to a common misconception, the relative amounts of rolling and sliding contact cannot be modified. He refers to tests with the Daimler-Lanchester dynamometer, in which worm gears, with which there is only sliding friction, showed surprisingly high efficiencies.

This introduces a discussion of a type of abrasion observed on high-duty spur gears, which points to a conclusion opposed to the common belief. On examining a number of pairs of gears, where the commonly observed wear was in the initial stages only, he found that there was an area near the pitch-line where the surface had become concave. This area always occurred, not at the pitch-line, but slightly toward the tip of the tooth, so that the worn areas did not come together on the mating gears, but the shoulder of each wore into the other.

The author presents the explanation that this can result from actual metallic contact and abrasion between the gears at portions of the surface where sliding contact occurs at a very low rate of speed. He points out that this type of wear is particularly productive of noise, as it introduces decided irregularities in angular velocity. Since noise seems to be produced even by perfect gears, he suggests that it may result from the reversal of stress, as the sliding contact becomes rolling contact and then reverses its direction, thus giving rise to a definite periodic force that necessarily must be transmitted to the gears and shafts, and may produce resonance effects in any of these parts that have a similar natural period of vibration.

Two possible remedies for the observed type of wear are suggested. One is the adaptation of types of gearing in which there is only sliding contact. This does not seem practicable at present, on account of size limitation for transmission gears and the like. The other suggestion is the adoption of metals that show less tendency to adhere in contact. Metals of dissimilar character usually behave better in this respect and, while the author sees no likelihood of using dissimilar metals for transmission gears, he suggests that much might be gained by a study of the behavior of dissimilar steels for this purpose. He does not take up the subject of the quality of lubricant as affecting the problem, but this too might be a fruitful field for further research.

Some topics that call for further study in connection with the relation of gear wear to gear life are

(1) Gear material; hardness and structure
(2) Tooth form; radius of curvature and unit pressure
(3) Peripheral speeds; as they affect both the tooth pressures and the impact and rubbing velocities
(4) Lubrication; effect of lubricants on gear wear, and the theory of gear lubrication
(5) Abrasive wear; the effects of the usual forms of foreign matter are little known

The causes of noise and the question of gear efficiency are left for discussion at a later date.

*(To be concluded)*

# MUTUAL BENEFITS OF FOREIGN TRADE[1]

BEFORE the war a great international system of beneficial trade had been built up, gradually and naturally; we scarcely realized how. The industries of the world had become in large degree interdependent. Western Europe was densely populated and highly industrialized. It was the focus of the world's exchanges. It was constantly sending out great quantities of manufactured goods in exchange for raw materials and foodstuffs. The war demoralized the industries of Europe and broke down the system of exchanges. Even within Europe the old channels of trade have been blockaded. The old Austro-Hungarian Empire, within which trade formerly was free and unrestricted, was divided into six independent states, each of which proceeded forthwith to erect high customs barriers against the others. Russia was formerly a great source of foodstuffs and raw materials for Western Europe, for which payment was made in manufactures; that trade has disappeared. The relations between Europe and the rest of the world have been interrupted in the same manner. India is a great tea-producing country, and Russia was one of the chief markets for tea. The inability of Russia to take its usual quantity of tea has prostrated the tea industry of India, and the inability of India to sell her products has cut down her purchases of cotton goods in England, and finally the inability of England to sell cotton goods has reduced her purchases of raw cotton in the United States. And so in this and other ways the European situation reacts upon us.

Our territory is so extensive and our resources so varied that we are in better position to live within ourselves than any other country. If our industries had been developed with that in view we might have been able to get along within ourselves, not completely, but more fully than at present. But our industries have been developed as a part of the industrial organization of the world. Although our exports are but a small part of our aggregate production, they represent great industries in which many millions of our people are engaged; and the price obtained for the portion that is exported affects the price for the whole production. Cotton mills of the United States have capacity to work up only about one-half of our cotton crop; the remainder must be exported. The lands that are growing cotton cannot be shifted to other crops without over-production of those crops, for we are exporters of all the principal agricultural products. In short, it is not practicable to shift the population into the new industries that would be necessary for the United States to live within itself. Furthermore, it is safe to say that no such policy would ever satisfy the ambitions of the American people or would be long maintained even if adopted.

A state of trade with balances running continually one way is abnormal and cannot possibly be permanent, because the debtor country cannot find the means of payment. Its efforts to procure the means of payment are certain to send exchange to a premium, and the premium will rise until it puts a check upon purchases in the creditor country and so brings the trade back into balance. Exchange rates act as an automatic governor. We are accustomed to say that the exchanges are in our favor when the dollar rates above other currencies; and they are in our favor for buying purposes, but not for selling. They help us to make purchases in other countries, but they do not help us to sell in other countries. On the contrary, they are a handicap upon sales.

The fact that international payments must be made for the most part in goods is very well illustrated by the difficulties that arise in the case of the German reparations payments. Most of us would like to see Germany pay, as far as is humanly possible, for the damages done by her armies and on the sea during the war; but many people do not understand the difficulty about paying great sums in another country and in another money. The reparations payments must be made in gold or in goods, and Germany has but little gold. The payments cannot be made in German paper money, for that is worthless outside of Germany and not worth much in Germany. If you were to visit Germany and see the many signs of wealth in the country, the fine cities, the great industries, the railroads, the forests and mines and farms, you might conclude that Germany was able to pay large sums; but none of these kinds of property can be transported out of Germany.

We have almost the same problem in the indebtedness running from the foreign governments to the United States Government. Some of our people are eager to have the payments begin; it has been proposed that the payments as received be applied to the payment of adjusted compensation to our ex-soldiers of the war. The people who urge this policy are apparently thinking that the payments will be made in money; but the payments cannot be made in money. The debtor countries are all collecting their taxes in depreciated paper currency; and even if they were able to collect from their taxpayers the sums nominally sufficient to make the payments to the United States, they could not make them in their paper currencies, because we have no use for those currencies. Those countries must export goods, which will create credits against which they must draw; and the very congressmen who have been urging that pressure be exerted upon the foreign governments to hasten payments, have been engaged at the same time in the preparation of a new customs tariff designed to reduce the offerings of foreign goods within this country.

As good merchants we are bound to give some attention to the means at the command of our customers for making payments. It is not enough that other countries shall want our products; they must be able to pay for them. The problem of payment is just as much ours as theirs, because they cannot pay without our help. The purchasing power of every

[1] From an address delivered before the Export Managers' Club of New York by George E. Roberts, vice-president of the National City Bank, New York City.

*(Concluded on page 106)*

# Camber and Gather Relationships in Front-Wheel Alignment

*By* J. C. Sproull[1] *Illustrated with* Drawings

**DEFINING the word "camber" to mean the tilting of the wheel spindles to make the wheels lean outward at the top, and the word "gather" to mean a second tilting of the spindles so as to bring the forward part of the wheels nearer together than the rear part, the author analyzes the effects produced by each of these operations and deduces mathematical formulas from which their proper values can be determined.**

**Tabular data and drawings are used in connection with the discussion of the effects of camber and gather on tire wear, and the use of the formulas and tables is explained in detail for an average case in which *S*, the slant height of the cone of which the wheel is the base, is taken as being approximately 500 in. In regard to misalignment, the author states that if this amounts to as much as ⅜ in., the life of a front tire may be reduced by many thousand miles.**

THE wheels of ox-carts in olden times were "dished" for structural reasons. In order that the spokes between the hub and the road might stand perpendicularly to the road surface, the spindles were "cambered." This caused the wheels to lean outward at the top. It was discovered that both tractive force and tire wear of such a pair of wheels could be reduced materially by giving the wheels "gather"; that is, by again tilting the spindles so as to bring the forward part of the wheels nearer together than the rear part.

The front wheels of modern motor-cars have both camber and gather, although the wheels are of the artillery type without dish. Reasons for camber and gather of the front wheels of motor cars are given in books of instruction published by manufacturers, and elsewhere; and are well understood by motorists generally. While the reasons for camber and gather in the front wheels of motor cars are altogether different from the reason, mentioned above, for cambering the axle of an ox-cart, the relation existing between camber and gather is similar. It is the purpose of this article to discuss this relation.

## Analysis

A wheel that leans outward at the top, that is, a combered wheel, can be considered as the base of a cone lying upon its side as shown in Fig. 1. The "natural" path of such a wheel, therefore, when it rolls on a plane surface, is the circumference of a circle described about the apex of the cone as a center. Obviously, unless the front wheels have gather, there will be a tendency for them to separate when they roll on a plane surface; this will result in a mutual slight side-skidding of both wheels, the right-hand wheel skidding toward the left and the left-hand wheel toward the right.

It is seen that this constant side-skidding, even though comparatively slight, will, if continued indefinitely, prove to be destructive to both tire and road surface. Thus, if the tendency of a wheel to turn outward is represented by an angle of only 0.2 deg., which corresponds to a misalignment of ⅛ in. for a 35-in. wheel, or an error of ¼-in. in gather, the effect in traveling 1000 miles will be equivalent to dragging the tire slowly sidewise under load a distance of more than 3 miles. If, however, the road surface at the point of contact with the tire is normal to the plane of the wheel, as might be the case when traveling upon a crowned road, then the wheel must be considered as a right section of a cylinder and its natural path will be a straight line. Therefore, the wheels should have no gather for this special case, even though they have camber. Where the road crown is excessive so that the angle $a$ is less than angle $b$ in Fig. 2, the gather becomes negative; that is, the wheels should "toe-out" rather than "toe in." Such a condition is extremely rare. The conclusion is that, for a given camber, gather should be less for vehicles that habitually travel over highly crowned roads than for those that commonly travel on comparatively flat roads.

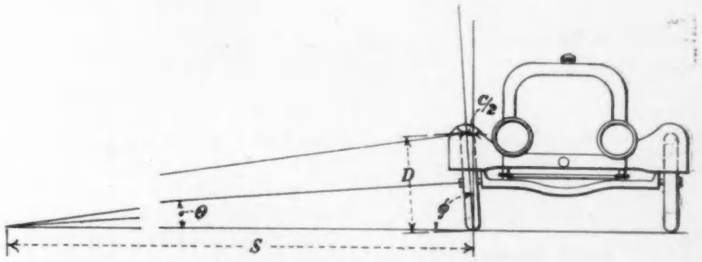



Fig. 1—Diagram Showing How a Cambered Wheel Resembles the Base of a Cone Lying on Its Side

Consider the case of a flat road and let

$2\alpha$ = Angle between two lines in the plane of the road intersecting at the apex of the cone and touching opposite ends of the area of contact between the tire and the road as shown in Fig. 3

$\beta$ = Angle of gather for one wheel, in degrees

$C$ = Wheel camber, equal to the difference of the distances between the tops and the bottoms of the vertical diameters of the tires, in inches

$D$ = Diameter of the wheel, in inches

$G$ = Gather, equal to the difference of the distances between the rears and the fronts of the horizontal diameters of the tires, in inches

$L$ = Length of area of contact between the tire and the road, in inches

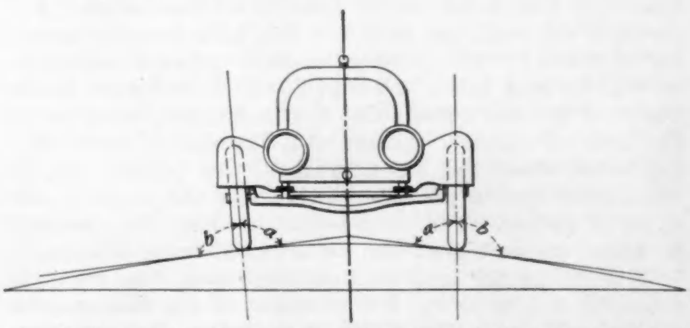



Fig. 2—Conditions That Exist Where the Crown of the Road Is Excessive

[1] Engineer of tests, B. F. Goodrich Co., Akron, Ohio.

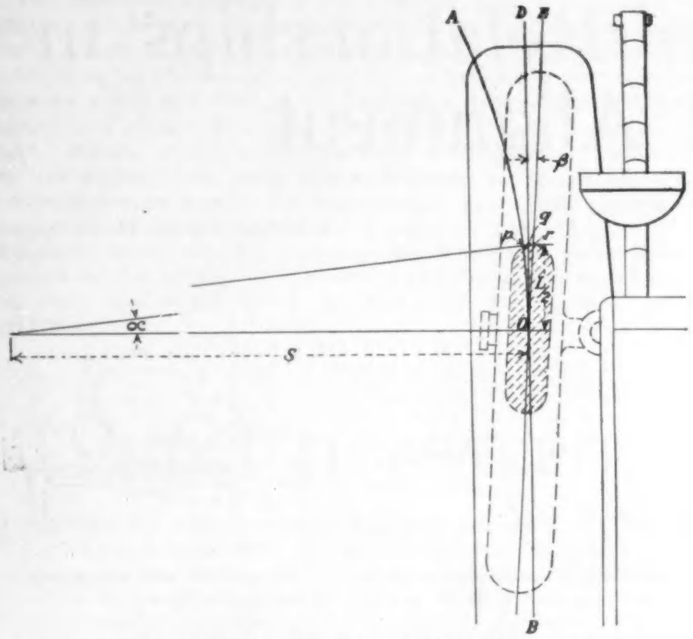



FIG. 3—DIAGRAM ILLUSTRATING THE RELATIONS THAT EXIST BETWEEN CAMBER, GATHER AND TIRE WEAR

$\Phi$ = Angle between the planes of the wheels and the road, in degrees
$S$ = Slant height in inches of the rolling cone of which the wheel is the base, as shown in Fig. 1
$\Theta$ = Spindle camber, equal to one-half the apex angle, in degrees, of the rolling cone as shown in Fig. 1

Then, according to Fig. 1,

$$\cos\Phi = \sin\Theta = C/2D \quad (1)$$
$$S = \tfrac{1}{2}D\sin\Theta = (D \times 2 \times D)/2C = D^2/C \quad (2)$$

An examination of a number of representative motor vehicles indicates that the value of $S$ can be taken as 500 in. In other words, the spindle camber $\Theta$ has been established by custom as approximately 2 deg. Accordingly we can write the simple relation $C = D^2/500$, which approximates the average case closely, and from which relation we compute the values shown in Table 1.

TABLE 1—CAMBER FOR VARIOUS WHEEL DIAMETERS

| $D$, Wheel Diameter, in. | $C$, Camber, in. |
|---|---|
| 30 | 1.80 |
| 32 | 2.12 |
| 34 | 2.31 |
| 36 | 2.59 |
| 38 | 2.88 |
| 40 | 3.20 |
| 42 | 3.52 |

## CAMBER, GATHER, AND TIRE WEAR

To get a clear conception of the relation existing between camber, gather and tire wear, it is necessary to take into consideration the area of contact between the tire and the road, the length of this area being of greatest moment. This at once involves inflation pressure, as will be seen later. In Fig. 3, which is drawn in the plane of the road, the point $O$ represents the center of the area of contact; the arc $OA$, the natural path of a cambered wheel that has no gather; the line $BD$, the direction of motion of the vehicle; and the angle $\beta$, the angle of gather. The slant height of the rolling cone is $S$, which we have seen can be assumed to be 500 in.

Without gather and without restraint, the point $O$ presently will be at $P$. The restraint of the axle and the spindle will cause the point $O$ to follow the path $Oq$, gather or no gather. Restraint exerted by the axle means tire destruction, since the restraining force must be transmitted to the road through the tire. If now we direct the wheel along the line $OE$ so that the distance $qr$ equals the distance $Pq$, then, for the region covered by the area of contact, no external restraint will be required since these two opposite tendencies are equal and their resultant is zero. Our problem then is reduced to the simple requirement of making $Pq = qr$. We have,

$$\tan\beta = (1 - \cos\alpha)/\sin\alpha \quad (3)$$
$$\sin\alpha = \tfrac{1}{2}L/500 = L/1000 \quad (4)$$
$$\cos\alpha = \sqrt{(1{,}000{,}000 - L^2)/1{,}000{,}000} \quad (5)$$
$$\beta = \tan^{-1}[\{1000 - \sqrt{(1{,}000{,}000 - L^2)}\}/L] \quad (6)$$

Equation (6) is for values of $L$ from 6 in. to 10 in., very nearly,

$$\beta = \tan^{-1} L/2000 \quad (7)$$

Solution of equation (7) gives values for $L$ and $\tan\beta$ as shown in Table 2.

TABLE 2—VALUES OF $L$ AND TAN $\beta$

| $L$, in. | Tan $\beta$ |
|---|---|
| 6 | 0.0030 |
| 7 | 0.0035 |
| 8 | 0.0040 |
| 9 | 0.0045 |
| 10 | 0.0050 |

Since the sine and the tangent are practically equal for the very small angle given in Table 2, we can also write

$$\tan\beta = \tfrac{1}{2}G/D = G/2D \quad (8)$$

Hence

$$G = 2D\tan\beta$$

and we have the relations for $L$ and $G$ shown in Table 3.

TABLE 3—VALUES OF $L$ AND $G$

| $L$, in. | $G$ |
|---|---|
| 6 | 0.006$D$ |
| 7 | 0.007$D$ |
| 8 | 0.008$D$ |
| 9 | 0.009$D$ |
| 10 | 0.010$D$ |

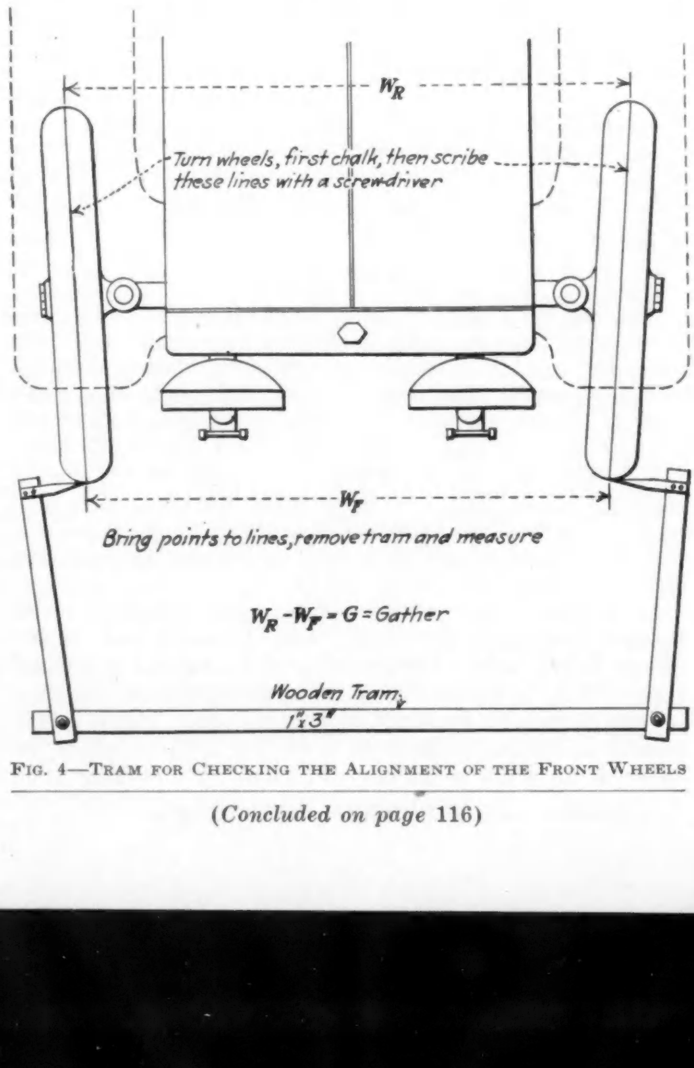



FIG. 4—TRAM FOR CHECKING THE ALIGNMENT OF THE FRONT WHEELS

(Concluded on page 116)

# Variations in Modern Diesel-Engine Design

*By* THOMAS ORCHARD LISLE[1]

PENNSYLVANIA SECTION MEETING *Illustrated with* PHOTOGRAPHS AND DRAWINGS

SUBSEQUENT to an outline of the great number of different designs of Diesel engine and an indication of the many variations in their mechanical features, the author comments unfavorably upon the tendency of builders of Diesel engines to depart from the original design unnecessarily and believes the time has come to restrict this practice.

The illustrations include a number of the leading types of American and European marine Diesel engine and brief comment accompanies each one. The different modes of departure from original design are indicated. Combination and opposed-piston engines. mechanical-injection and double-acting Diesels, compound engines and engines representing the latest European practice are shown and described. Brief comment is made also on the subject of Diesel-engine fuel-consumption.

A SHIP owner who is not familiar with the motorships of today, and seriously considers the adoption of economical oil-engine power for his fleet for the first time, is confronted by an almost bewildering variety of Diesel engines from which he must make his selection, many of which differ radically in design from each other. All have certain advantageous claims, but there is little or no semblance of standardization in either framework construction or in methods of combustion.

There are more than 50 different designs of Diesel engine on the market. The designs include

Two cycle, with
- Air-injection
- Airless injection
- Scavenge valves
- No scavenge valves
- Combined scavenging and injection valves
- Scavenging pumps driven by cranks at the end of the engine
- Scavenging pumps operated by rocking levers from the crossheads
- Stepped scavenging pistons
- Separate electrically driven scavenging blowers

Four cycle, with
- Air-injection
- Airless injection
- Vertical inlet and exhaust-valves
- Horizontal valves

Variations, inclusive of
- Trunk-piston
- Crosshead
- Low compression, with auxiliary combustion-starting system
- Medium compression, with or without steam-heated cylinders
- Medium compression with electric or red-hot point starting devices
- Diesel combustion on one side and steam pressure on the other side of the piston
- Single-acting
- Double-acting
- Opposed piston, and variations of this particular design
- Use of cylinder liners
- Non-use of cylinder liners
- Detachable cylinder-heads
- Cylinder-heads cast integrally with the cylinders
- Heavy cast-iron frames
- Steel columns and no cast-iron frame
- Both frames and columns of cast iron
- Cylinders cast separately
- Cylinders cast in-block
- V-type cylinders
- Miscellaneous, some of which can be called Diesel engines only by stretching a technical point

All of these have variations in their reversing mechanisms. Some have fresh or salt-water-cooled pistons, others have oil-cooled pistons and some have no cooling medium other than the atmosphere. Lastly, there are advocates of the Diesel-electric drive; and of the Diesel reduction-gear drive, as opposed to direct drive.

Nearly every engineering company that recently has turned its attention to Diesel-engine construction has seemed to desire to produce an engine that is unlike anything previously invented, rather than to follow established successful practice conservatively, within reason but without unnecessary "plagiarism," or else to adopt a careful combination of the best designs in a businesslike manner. It generally is better to pay a royalty for a known good product, if necessary, than to try indirectly to obtain something that is only partly as successful and as practical commercially.

Is it any wonder that some ship owners acknowledge their uncertainty and turn in desperation to geared turbines and turbine-electric drives, enduring the consequent particular troubles and anti-economies that are attendant upon them? There appears to be an almost innumerable number of new designs steadily coming on the market; therefore, is it not time to call a temporary halt? Ultimately, of course, some designs will be eliminated, as has been true of other previous ones.

In the face of following this zig-zag course, with submerged dangers lurking ahead, to port and to starboard, the Diesel engines that have been adopted more extensively than any other type of engine for large merchant ships have been of simple and straightforward design and follow recognized steam-engine design as nearly as is practicable. On the other hand, a number of engines that differ radically have given excellent results in the limited number of vessels in which they have been installed; so, it is exceedingly difficult to draw a definite line of development, because the different models may give equally good results if well constructed. The illustrations show a number of the leading types of American

[1] Editor of *Motorship*, New York City.

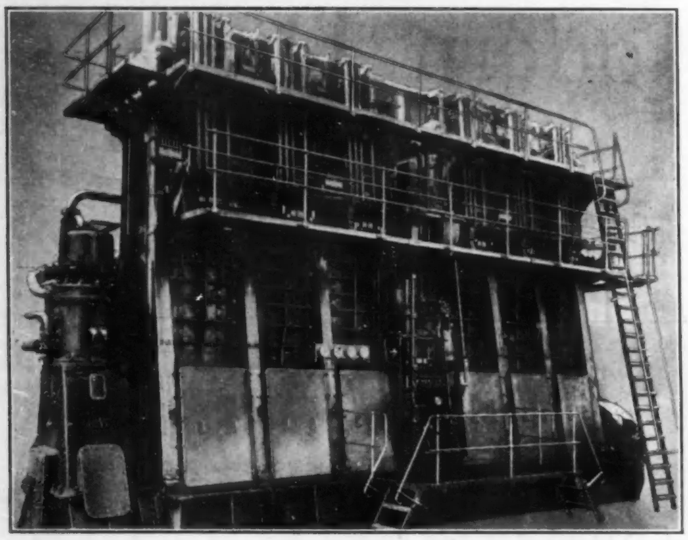


FIG. 1—THE BURMEISTER & WAIN LONG-STROKE DIESEL ENGINE

and European marine Diesel engine, and I will comment briefly upon each design.

Fig. 1 shows the Burmeister & Wain long-stroke design, which is one of the best known. Seven companies, located in Sweden, Norway, Denmark, Scotland, Germany and America build this engine under license, and all have made their constructions with practically no change in the design. This engine is of very robust and straightforward construction without having had any attempt made toward radical departure from known marine practice. I consider that this is responsible for its success, in conjunction with sound engineering. During the 10 years since the first Burmeister & Wain engine was completed, the design has not been without its numerous slight troubles due to minor parts but, although nearly all of these have been eliminated by modification, the present model is almost the same in appearance as the original design. The Cramp shipyard in Philadelphia is building four such 2250-i.hp. engines.

Fig. 2 shows an American version of the same engine, the Worthington, although it is not built under a license. This engine follows the earlier design more closely. It has cylinders that are cast separately, instead of the box type of cylinder cast three per set or in single boxes bolted together. Its main differentiation from the Burmeister & Wain design is that the front frames have their lower parts detachable to facilitate the removal of the crankshaft. There is also an auxiliary exhaust-port in the lower end of each cylinder that can be cut-out if desired.

Another well-known and successful marine Diesel engine is the Werkspoor engine shown in Fig. 3. It is

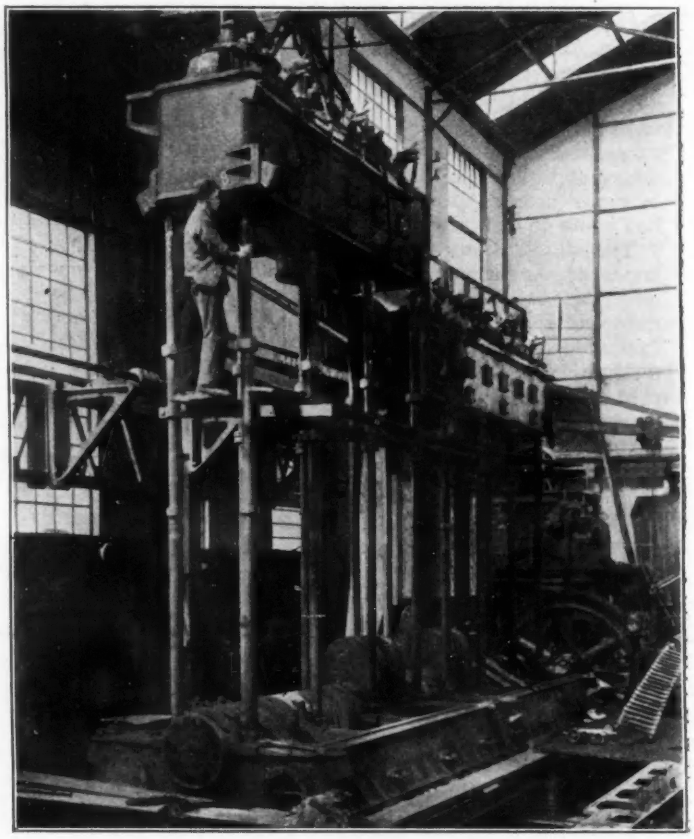


FIG. 3—THE WERKSPOOR MARINE DIESEL ENGINE IN COURSE OF CONSTRUCTION

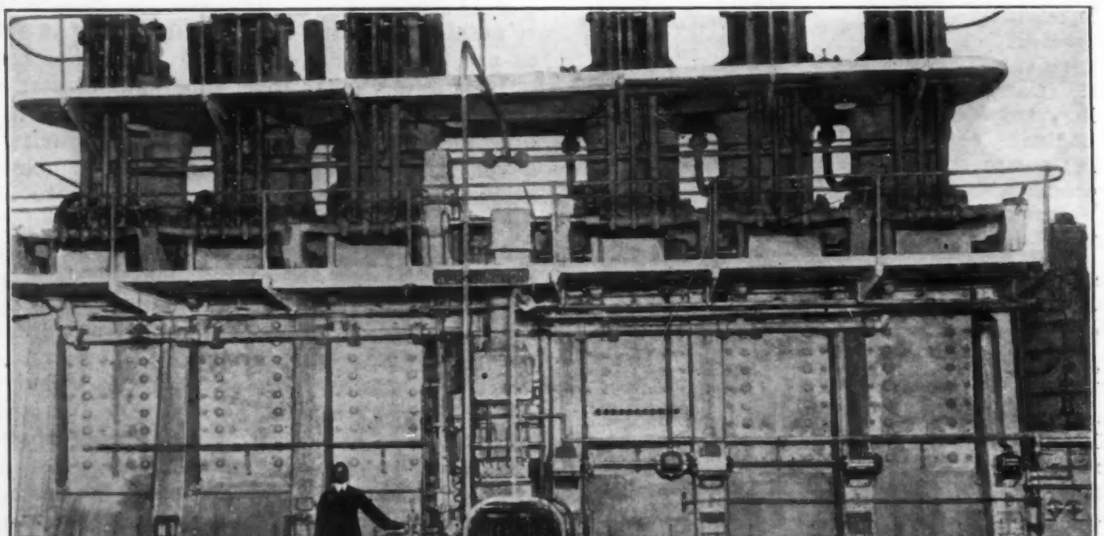


FIG. 2—THE WORTHINGTON, AN AMERICAN TYPE OF DIESEL ENGINE

constructed along lines that are very different from those of the Burmeister & Wain design and yet it follows recognized lines of marine steam-engine design. It, also, is of the four-cylinder single-acting type. Fig. 4 shows that the large cast-iron frames similar to those in the Burmeister & Wain engine are replaced by lighter steel columns, and that steel tierods run from the bedplate to the top of the cylinder box. The cylinders have no jackets in the ordinary sense or detachable heads, but are of mushroom construction and "dropped" into the box, the latter also acting as a lateral girder. The cylinders have detachable extensions for removing the pistons. The cast-iron columns at the back are used for carrying the crosshead guides and air intercoolers. These features of the construction are different from any other designs, although the cylinder-box system has been adopted by the Tosi, Krupp, North British, Holeby, Polar, Burmeister & Wain and several other companies in recent years. I believe it will come into more general practice later on. About 10 companies build the Werkspoor engine. Fig. 5 gives a better idea of the steel-column construction and shows more clearly the cast-iron columns at the back of the Werkspoor engine that are used for carrying the crosshead guides and intercoolers for the air-compressors. The diagonal tierods are not shown.

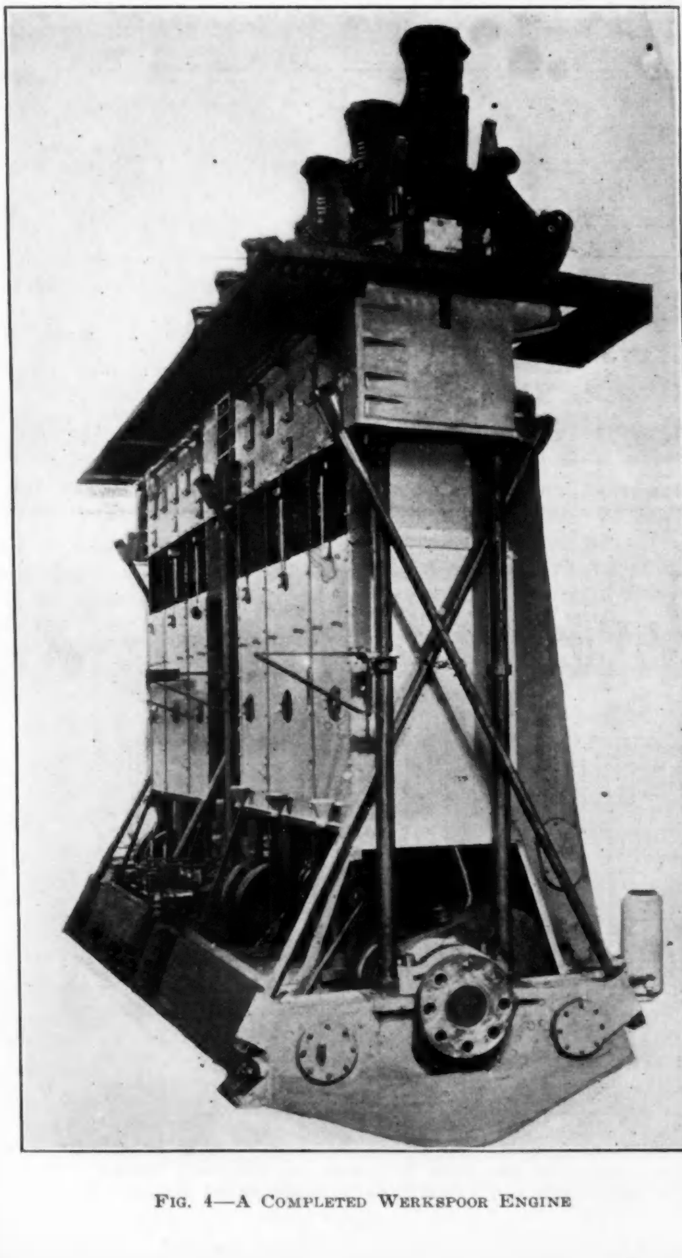


Fig. 4—A Completed Werkspoor Engine

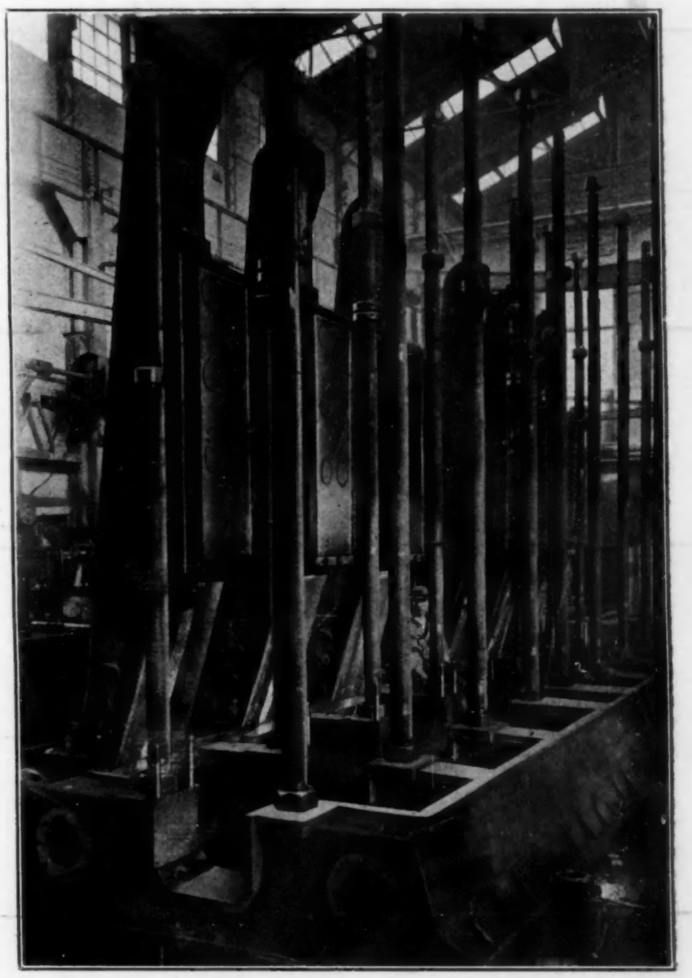


Fig. 5—In the Werkspoor Engine the Heavy Cast-Iron Frames Are Replaced by Lighter Steel Columns and Steel Tierods Extend from the Bedplate to the Top of the Cylinder Box

### Departure from Original Design

The question arises in regard to how far a licensee should depart from the original design to suit local conditions of construction and operation. Fig. 6 shows one of the Werkspoor engines constructed for the Standard Oil Co. tanker H. T. Harper by the Pacific Diesel Engine Co., Oakland, Cal. There are three firms building Werkspoor Diesel engines in the United States. All three are working along different lines and all have changed the parent design somewhat. It will be noted that the Werkspoor engine of the Pacific Diesel Engine Co. has retained the steel columns, but the large cast-iron columns at the back and the diagonal tierods are dispensed with. The company has introduced a cast-iron frame in their place, through which the steel columns run and, by cast-iron distance-pieces between the frames and the cylinder box, it has secured the effect of vertical tierods. However, somewhat similar construction was used by the Werkspoor company in the engines of the motorships Vulcanus and Sembilan about 12 years ago, which are still in service; it was also used in some of its stationary and submarine engines.

Fig. 7 illustrates a 2000-i.hp. Werkspoor engine now being built at the Camden plant of the New York Shipbuilding Corporation. It differs from both the Holland-Werkspoor and the Pacific-Werkspoor engines in that

cast-iron A-frames run from the lower sides of the cylinder boxes to the bedplate and instead of the steel columns and diagonal rods, or the combination frame-

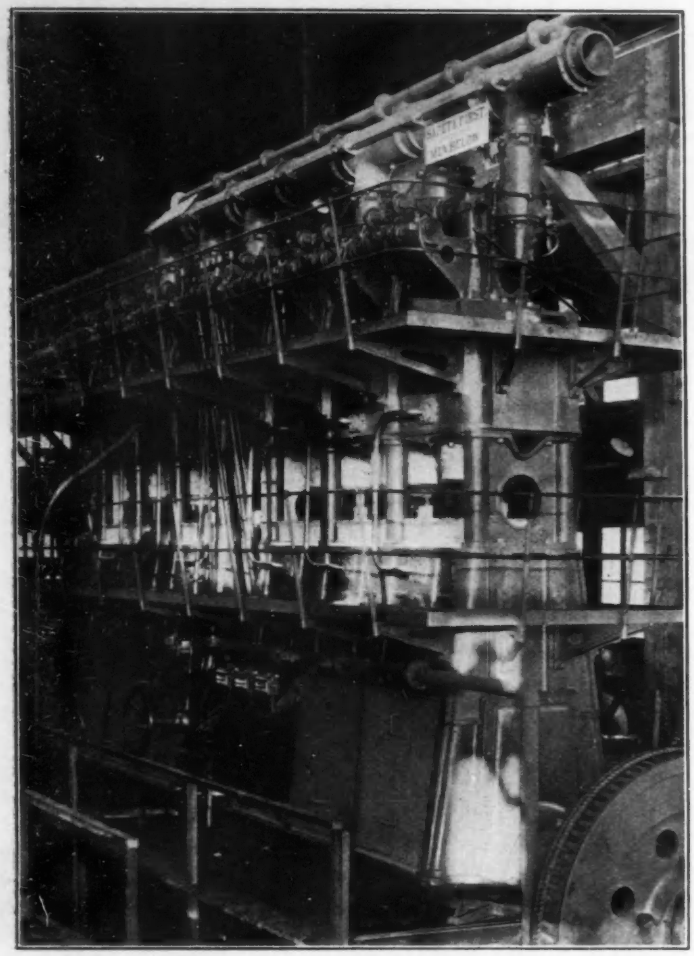


FIG. 6—A WERKSPOOR TYPE OF DIESEL ENGINE CONSTRUCTED ON THE PACIFIC COAST FOR AN OIL TANKSHIP

and-column arrangement. However, the upper part of the engine follows regular Werkspoor design. Another change is that the New York Shipbuilding Corporation

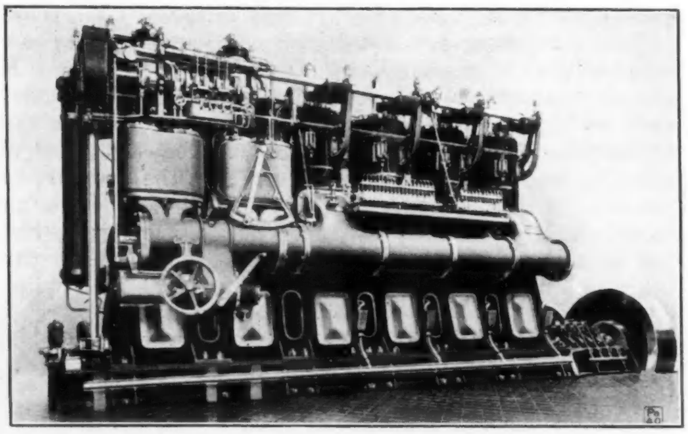


FIG. 8—A 360-B. HP. TRUNK-PISTON ENGINE OF THE TWO-CYCLE PORT SCAVENGING TYPE IN WHICH THE SCAVENGING CYLINDERS ARE ALSO EMPLOYED FOR AIR STARTING

has reverted to the stepped-piston type of air-compressor, instead of using the separate-stage compressor adopted by Werkspoor and others of its licensees.

Another case where licensors and their licensees make a number of distinctive types of engine is illustrated in Fig. 8 which depicts a 360-b.hp. trunk-piston type Polar Diesel engine of the two-cycle port-scavenging type in

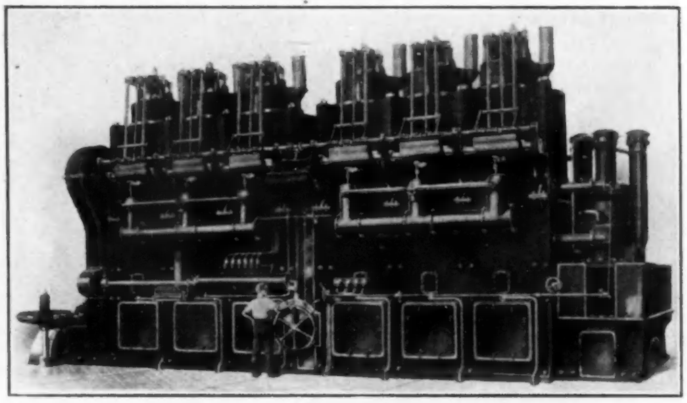


FIG. 9—A 900-B. HP. CROSSHEAD-TYPE ENGINE IN WHICH THE CYLINDERS ARE IN TWO BOX GROUPS AND BOLTED TO BOX-TYPE CAST-IRON FRAMES

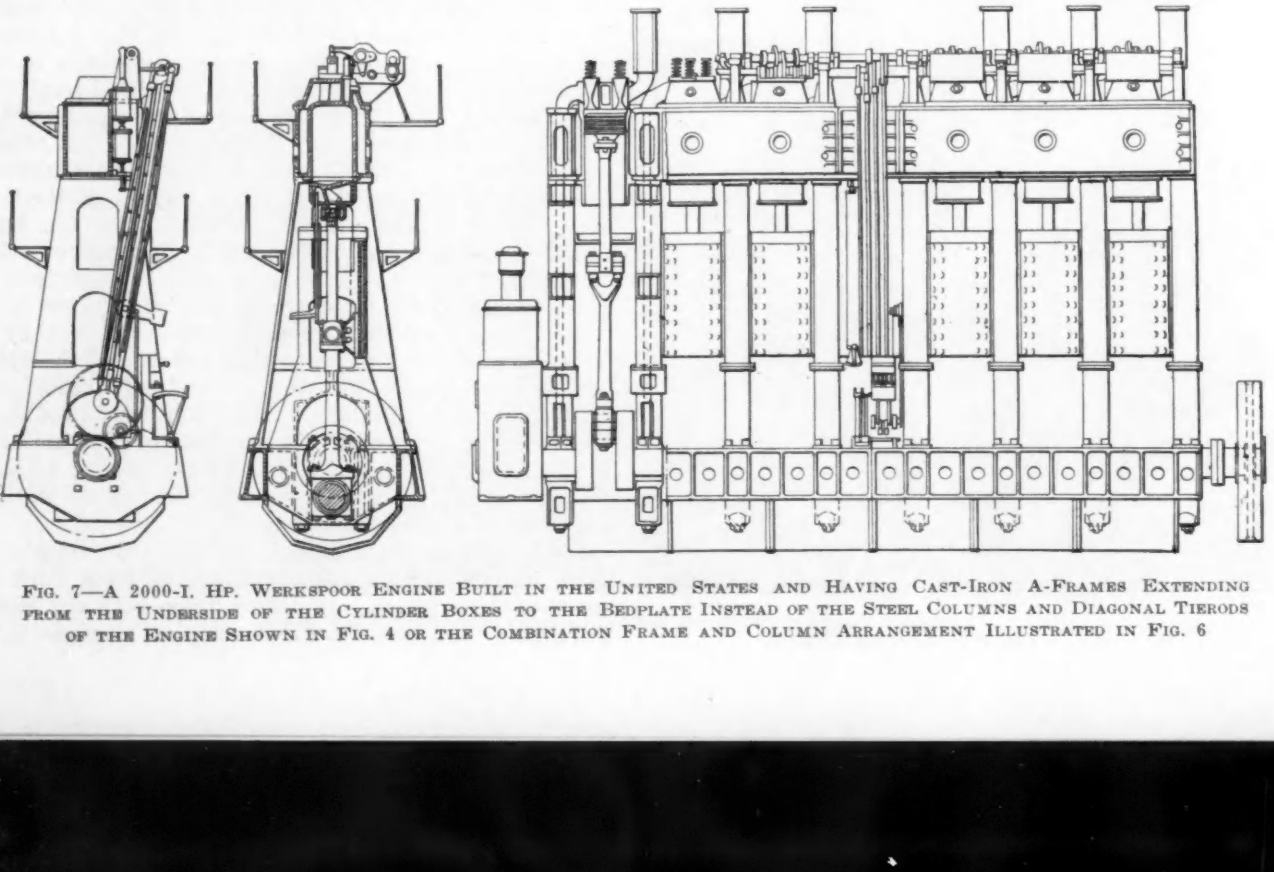


FIG. 7—A 2000-I. HP. WERKSPOOR ENGINE BUILT IN THE UNITED STATES AND HAVING CAST-IRON A-FRAMES EXTENDING FROM THE UNDERSIDE OF THE CYLINDER BOXES TO THE BEDPLATE INSTEAD OF THE STEEL COLUMNS AND DIAGONAL TIERODS OF THE ENGINE SHOWN IN FIG. 4 OR THE COMBINATION FRAME AND COLUMN ARRANGEMENT ILLUSTRATED IN FIG. 6

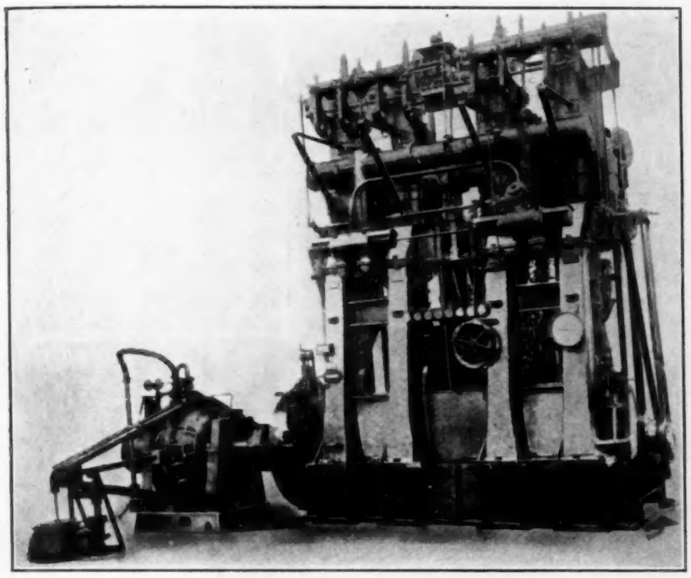


FIG. 10—A 600-B. HP. CROSSHEAD-TYPE ENGINE OF BRITISH CONSTRUCTION IN WHICH THE WORKING CYLINDERS ARE CAST SEPARATELY AND THE FRAMEWORK IS OF THE OPEN A-TYPE

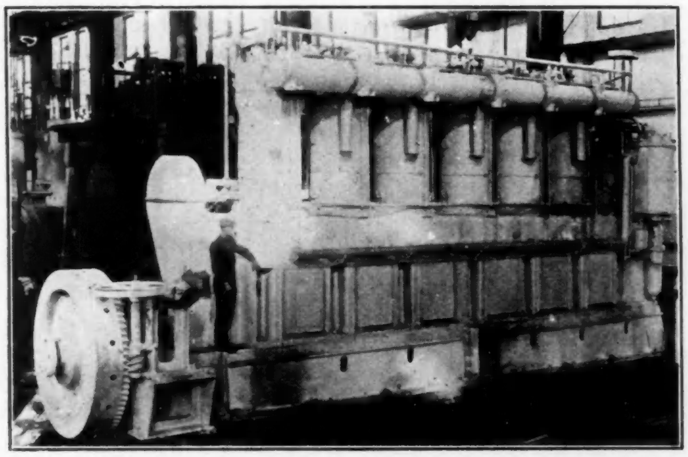


FIG. 11—AN AMERICAN TRUNK-PISTON TYPE 750-B. HP. ENGINE

which the scavenging cylinders are used also as air-starting cylinders. They can be seen at the forward end under the air-compressor stages. Fig. 9 shows a 900-b.hp. crosshead-type Polar engine designed for 135 r.p.m. that is built also by the Atlas Diesels Motorer and is of somewhat distinctive design. The cylinders are in two box groups, bolted on to box-type cast-iron frames. The cylinders for air-starting are below the working cylinders, compressed air being admitted to the under sides of the pistons by sliding valves. This is done to avoid the entrance of cold starting-air and its contact with the hot working piston. A somewhat similar ar-

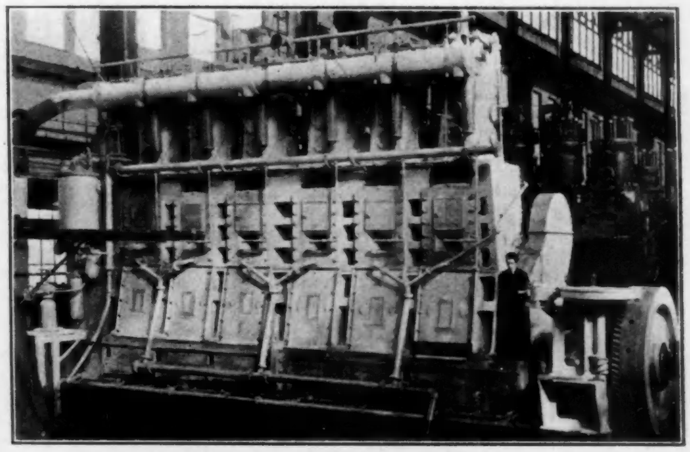


FIG. 12—A 900-B. HP. CROSSHEAD-TYPE AMERICAN ENGINE

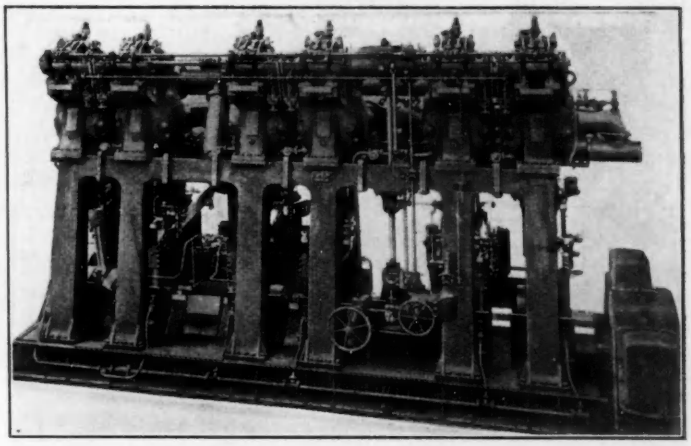


FIG. 13—A BELGIAN TWO-CYCLE CROSSHEAD-TYPE ENGINE EQUIPPED WITH VALVE-IN-HEAD CYLINDERS

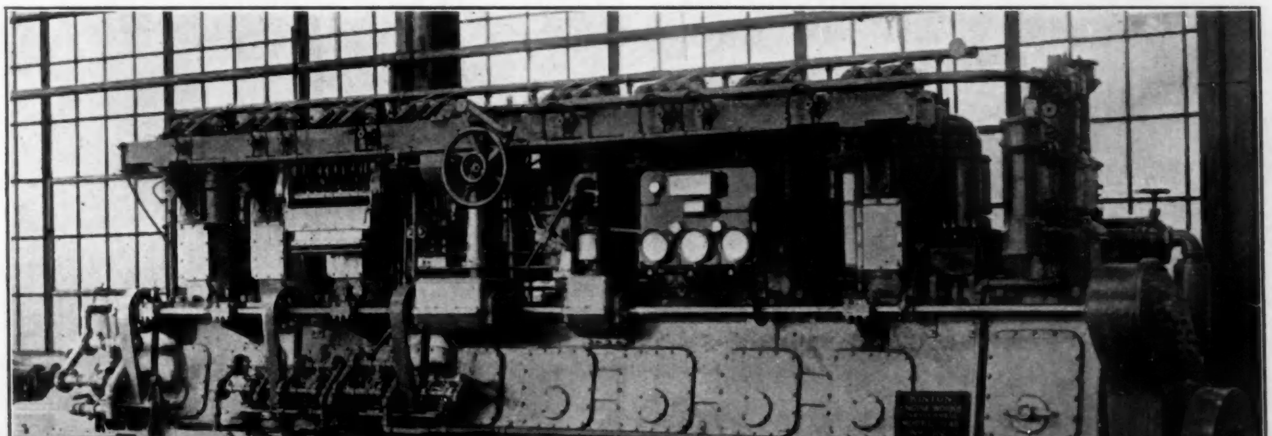


FIG. 14—A 400-B. HP. HIGH-SPEED TRUNK-PISTON FOUR-CYCLE ENGINE THAT IS USED EXTENSIVELY IN CONJUNCTION WITH THE ELECTRIC DRIVE

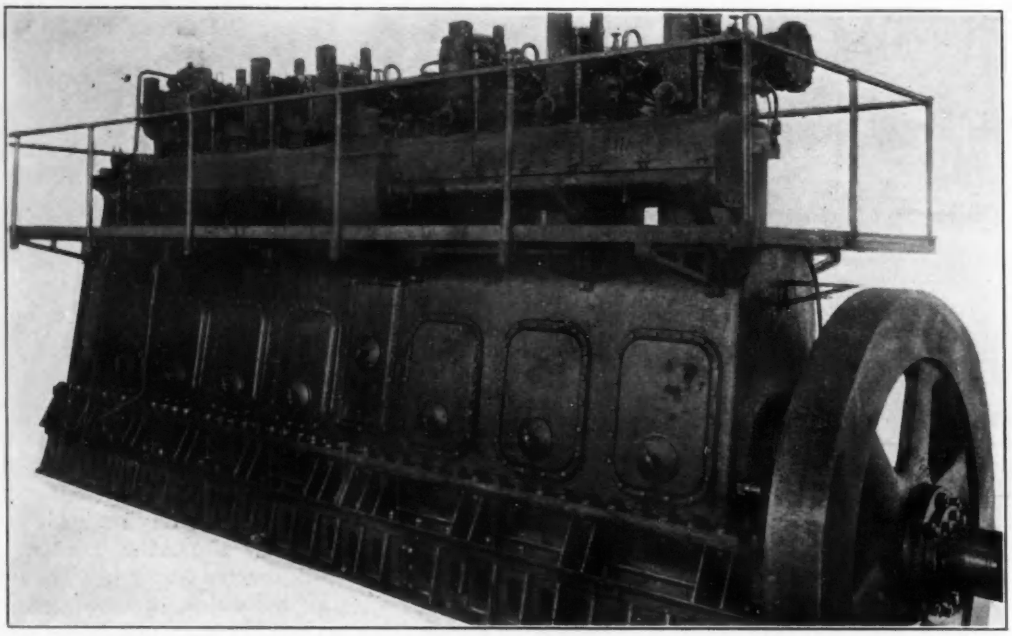


FIG. 15—A 700-B. HP. FOUR-CYCLE TRUNK-PISTON ENGINE HAVING THE INLET AND EXHAUST-VALVES HORIZONTAL INSTEAD OF VERTICAL

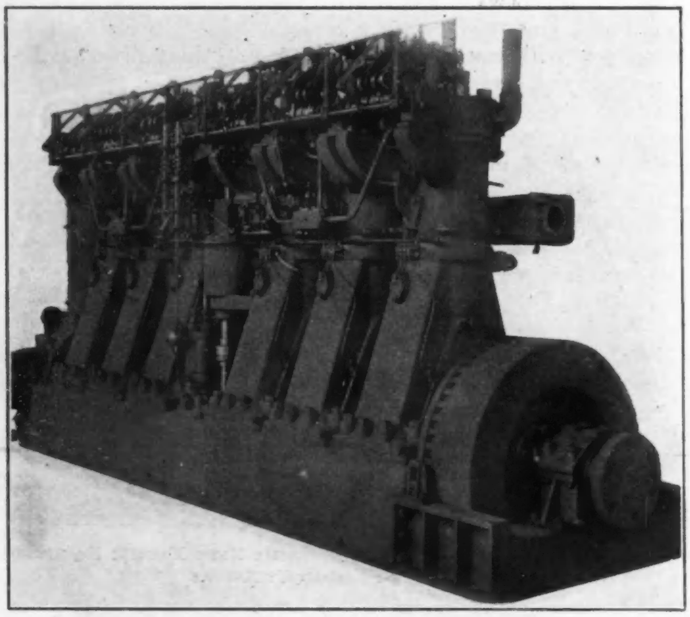


FIG. 16—AN AMERICAN TRUNK-PISTON FOUR-CYCLE ENGINE

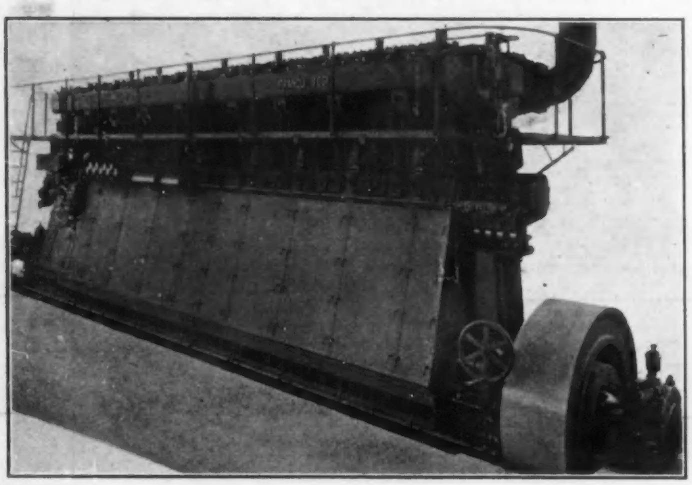


FIG. 17—AN ITALIAN FOUR-CYCLE CROSSHEAD-TYPE ENGINE THAT IS CHARACTERIZED BY SIMPLICITY OF DESIGN

rangement is shown in Fig. 10, which is a Neptune-Polar crosshead-type engine of 600-b.hp. that is built by the British licensee, Swan, Hunter & Wigham Richardson; but the general appearance of the engine is much different, it having working cylinders that are cast separately, and an open A-type framework.

Fig. 11 shows a trunk-piston type 750-b.hp. engine that is built by the American licensees, McIntosh & Seymour. Engines of this design and power have given splendid service in the Vacuum Oil Co.'s tankers Bayonne and Brammell Point. A crosshead-type 900-b. hp. engine is

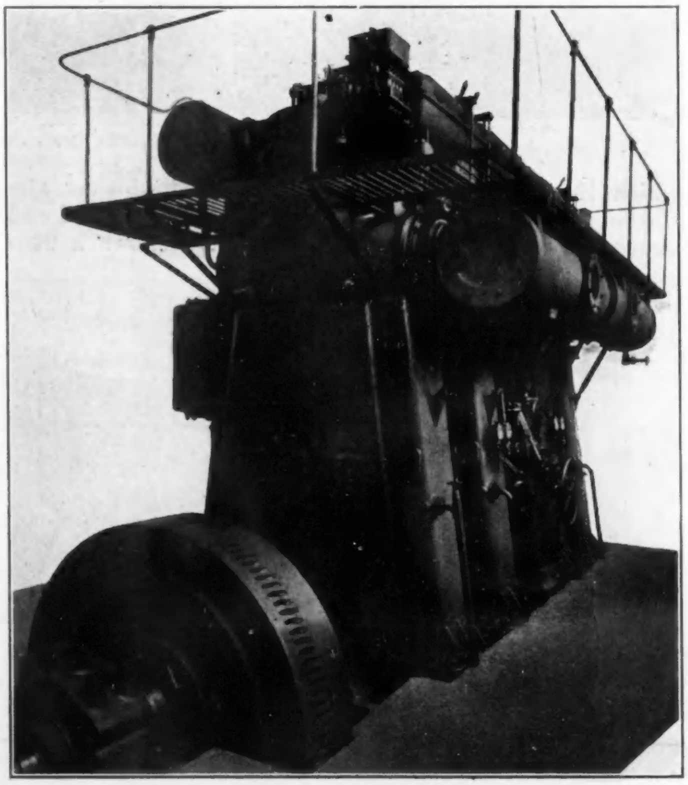


FIG. 18—A CROSSHEAD-TYPE TWO-CYCLE ENGINE OF ITALIAN CONSTRUCTION THAT WAS ABANDONED IN FAVOR OF THE FOUR-CYCLE MODEL ILLUSTRATED IN FIG. 17

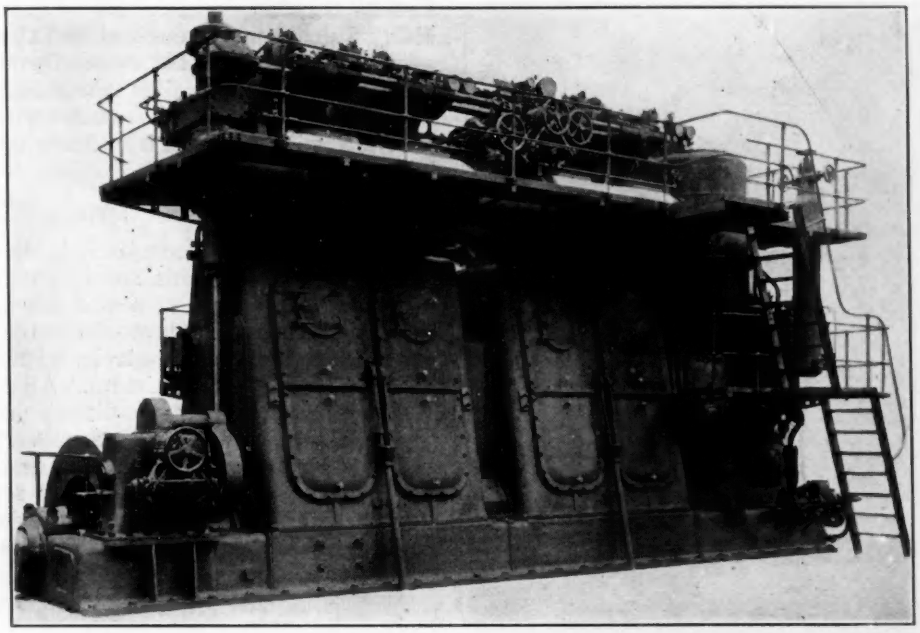


FIG. 19—A 1200-B. HP. PORT-SCAVENGING TYPE OF ENGINE HAVING THE SCAVENGING PUMP AND THE AIR-COMPRESSOR MOUNTED AT THE FORWARD END

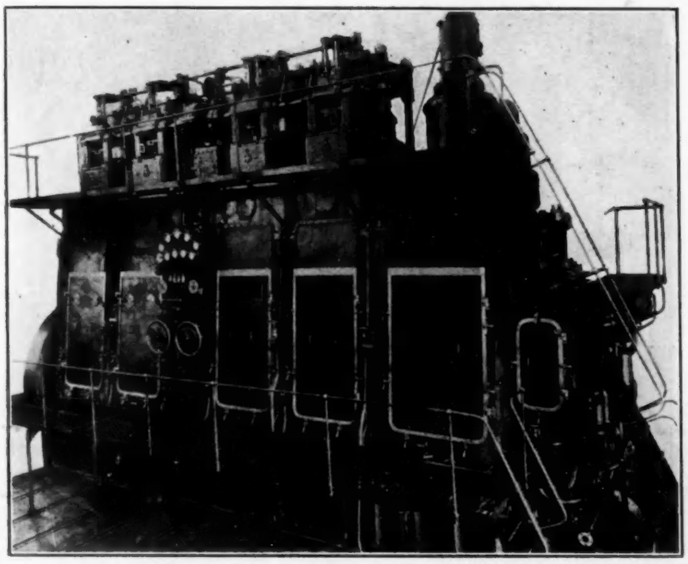


FIG. 20—A PORT-SCAVENGING TWO-CYCLE CROSSHEAD TYPE OF ENGINE IN WHICH THE CYLINDER-HEADS ARE BOLTED DOWN BY LONG TIERODS RUNNING DOWN TO THE BEDPLATE

illustrated in Fig. 12. This is installed in the American motorship Kennecott.

Fig. 13 shows a two-cycle crosshead-type engine of the valve-in-head scavenging type, having open cast-iron frames. It was built by Carel Fréres, of Belgium, and is constructed under license in America by the Nordberg Mfg. Co. of Milwaukee. The Schneider engine, which is constructed in France, is built along very similar lines. A much different type is illustrated in Fig. 14. This is the 400-b.hp. high-speed trunk-piston four-cycle Winton engine. It is built more along the lines of large marine gasoline engines than those of steam-engine practice and is used extensively in conjunction with electric drive, as well as with direct drive in vessels of moderate size. The United States War Department has installed 14 such engines in twin-screw concrete passenger-ships with great success. Another four-cycle trunk-piston engine, but one of lower speed, is the 700-b.hp. Nelseco engine shown in Fig. 15. This engine is original in that it has horizontal inlet and exhaust-valves instead of having the customary vertical valves.

A third type of American-built trunk-piston four-cycle marine Diesel engine is shown in Fig. 16. This is the Dow engine constructed under British license. There is only one American ship equipped with these engines and, although a wooden vessel, she has given splendid service.

For a simple design of the four-cycle crosshead-type engine, the Tosi Diesel engine shown in Fig. 17 has much to commend it. The Tosi firm also built a neat crosshead-type two-cycle Diesel engine such as is shown in Fig. 18, but this was not so successful and it was abandoned for the four-cycle model.

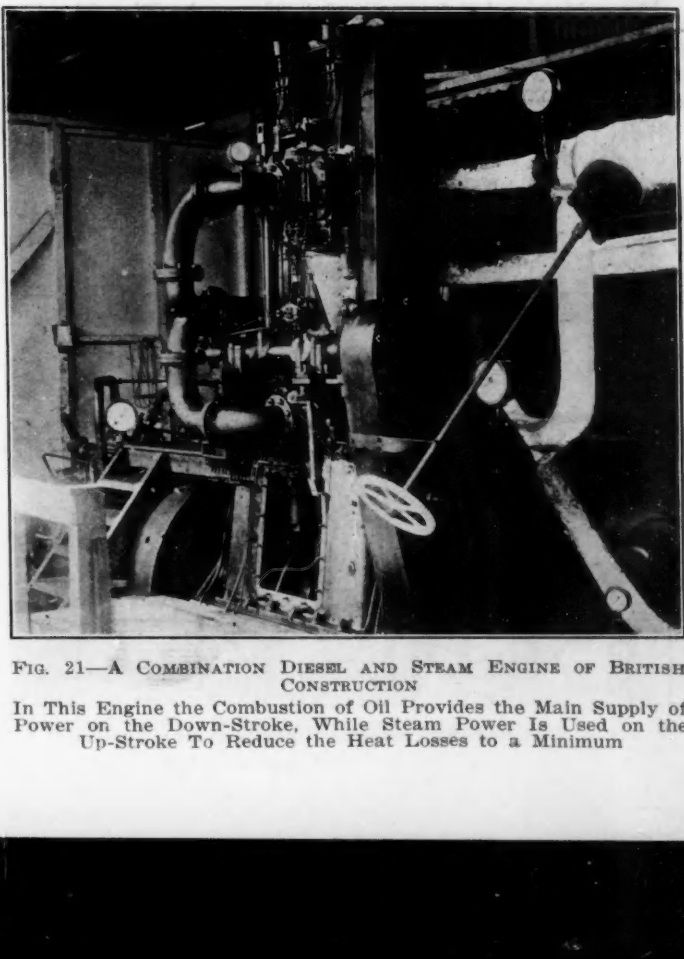


FIG. 21—A COMBINATION DIESEL AND STEAM ENGINE OF BRITISH CONSTRUCTION

In This Engine the Combustion of Oil Provides the Main Supply of Power on the Down-Stroke, While Steam Power Is Used on the Up-Stroke To Reduce the Heat Losses to a Minimum

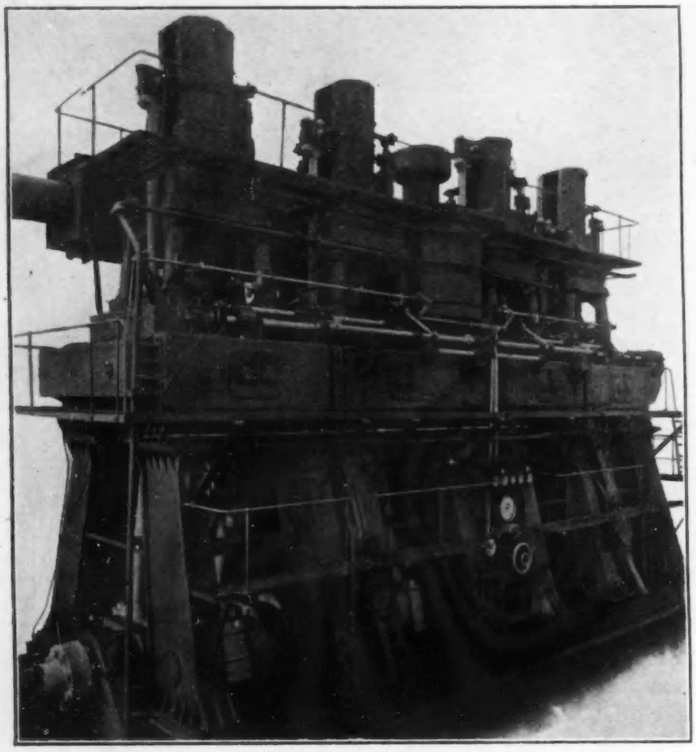


Fig. 22—A Four-Cylinder Crosshead-Type Opposed-Piston Two-Cycle 3000-I. Hp. Engine in Which the Pistons in Each Cylinder Move Apart and Combustion Takes Place between Them

Of all two-cycle crosshead-type Diesel engines, probably the simplest in design is the Sulzer engine, shown in Fig. 19, also built by the Busch-Sulzer Co., St. Louis. The model illustrated is a 1200-b.hp. engine of the port-scavenging class, and it is an exceptionally well-balanced job. The scavenging pump and air-compressor are at the forward end, but the scavenging pump is replaced by an electrically driven blower in new constructions. The Ansaldo-San Giorgio engine is of somewhat similar construction, yet different; it also is of the port-scavenging two-cycle crosshead type and is shown in Fig. 20. The method of bolting down the cylinder-heads by using long tierods running down to the bedplate should be noticed. Early Sulzer marine engines had this feature, but it was abandoned; the present Sulzer marine engines have no steel rods. Two smaller Ansaldo-San Giorgio engines were recently installed in an American freighter in conjunction with the electric drive of the General Electric Co.

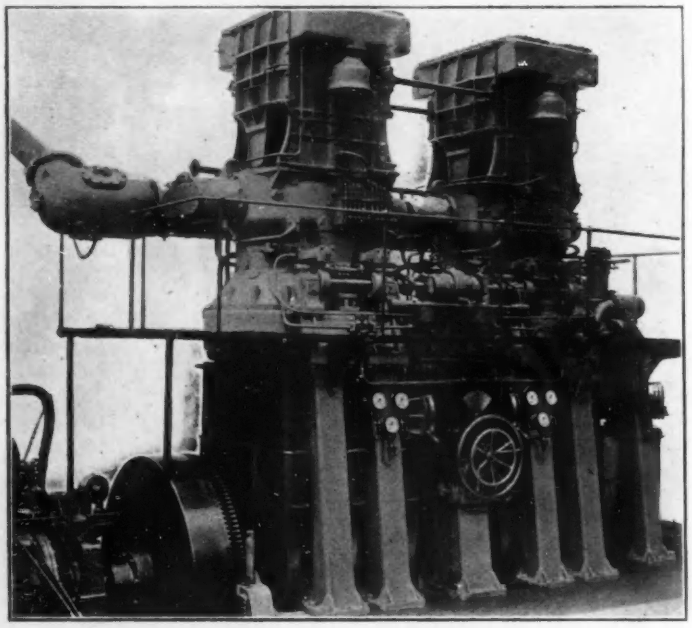


Fig. 23—Another Design of Opposed-Piston Engine

### Combination and Opposed-Piston Engines

Another radical departure is the Still combination Diesel-and-steam engine shown in Fig. 21. In this design the main source of power is secured from the combustion of oil on the down-stroke; supplementary steam power is used on the up-stroke with the object of reducing heat losses to a minimum. All the heat usually lost in the working-cylinder cooling-jackets is used for generating steam, as well as heat recovered from the exhaust gases by generators and feed-water jackets. Compounding in a single-cylinder engine is secured by using the lower ends of two combustion cylinders as high-pressure steam-cylinders and those of the other four working cylinders as low-pressure steam-cylinders. Overload is secured by oil-firing the steam generator. Oil-firing is used also for starting on steam. This furnishes initial heat to the Diesel cylinders and so enables the cylinder pressure to be kept down to 300 lb. per sq. in. as against 400 to 500 lb. per sq. in. for the straight Diesel engine. The single-cylinder unit illustrated develops 350 b.hp. at 350 r.p.m., or 540 b.hp. on overload.

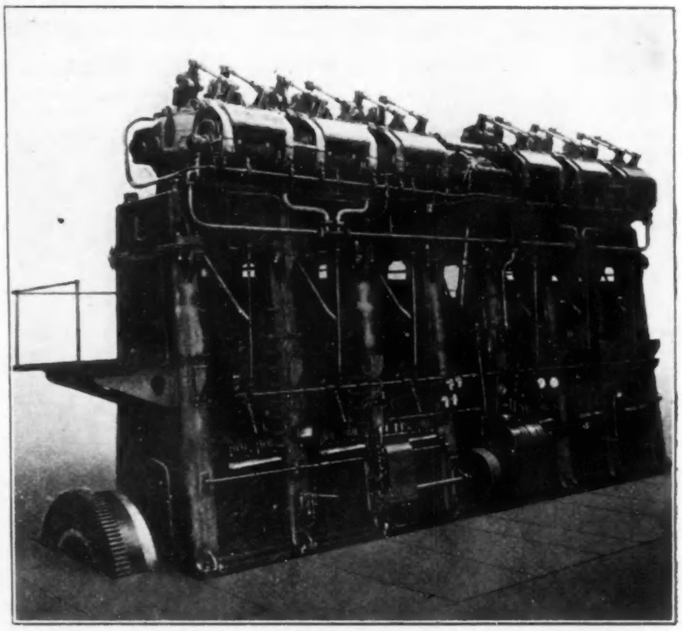


Fig. 24—A Four-Cycle Airless Injection Engine Developing 1250 B. Hp. at 115 R.P.M.

A large shipyard near Philadelphia is deeply interested in the opposed-piston type of Diesel engine. A Doxford four-cylinder crosshead-type opposed-piston two-cycle engine of 3000 i.hp. is illustrated in Fig. 22. The pistons in each cylinder move apart and combustion takes place between them on the Ochelhauser principle. The second vessel equipped with the Doxford engine was placed in service recently. Excellent results were obtained at the tests of both ships. Airless injection of fuel at high pressure has been adopted in conjunction with low cylinder-compression and, because of the latter, steam or hot water is used for warming the cylinder-jackets prior to starting from cold.

A different arrangement of the opposed-piston design is depicted in Fig. 23. This is the Cammell-Laird Fullagar marine Diesel engine, which, regardless of its radical departure from conventional design, is likely to come to the front, even if only because of the amount of engineering skill that is back of it. The firms having building licenses in England are John Brown & Co., Clydebank; Palmers Shipbuilding & Iron Co., Newcastle; Cammell-Laird & Co., Birkenhead; English Electric Co., London; Dunsmuir & Jackson, and David Rowan & Co., Glasgow. The Bretagne Shipbuilding Co., Nantes, holds a license in France. However, American firms do not as yet seem to have so much faith, although American engineers who have seen this engine have been very enthusiastic. It is an adaptation of the Ochelhauser system combined with bold defiance of the best engineering practice, which is to transmit force in a straight line so far as this is feasible. The upper piston in the Ochelhauser system is connected to a crosshead that carries two connecting-rods which pass to the sides of the cylinder to two crankpins outside and opposite to a central crankpin, to which the lower piston is connected by its rod, the three crankpins with their webs forming a single crank unit between a pair of bearings. In the Cammell-Laird Fullagar design, only a single crank is required per cylinder and two cylinders are required to form a complete unit. Instead of being an overhung arm placed over its crank in a fore-and-aft line, the crosshead of the upper piston is placed centrally in regard to its piston but in an athwartship line; and, instead of having a connecting-rod from each end directly down to its crank, there is a diagonal rod from each end of the crosshead to keep the strains central. These diagonal rods lead down to a crosshead from which a connecting-rod is attached to a second crank, the guide shoes being arranged to take the thrust of both the connecting and the diagonal rods. Thus, the upper piston of one cylinder is connected to the lower piston of the adjoining cylinder; and the lower piston is connected to the upper piston of the same adjoining cylinder. Another radical change in design exhibited by this engine is that the scavenging pistons and cylinders are of square section.

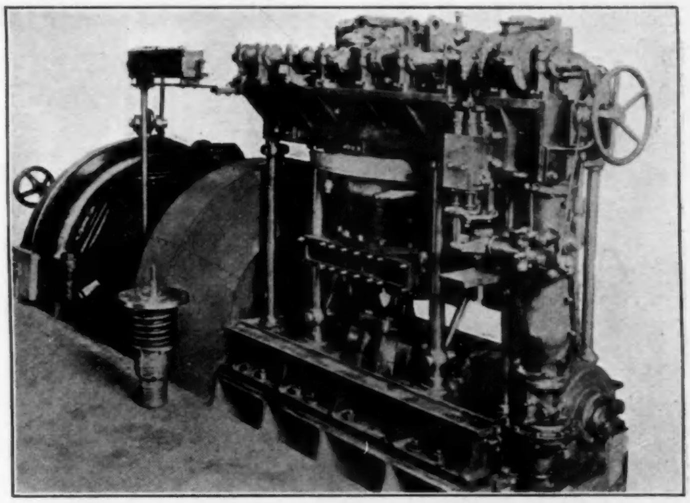


Fig. 26—An American Heavy-Duty Compound Engine Designed To Operate on Bunker Oil

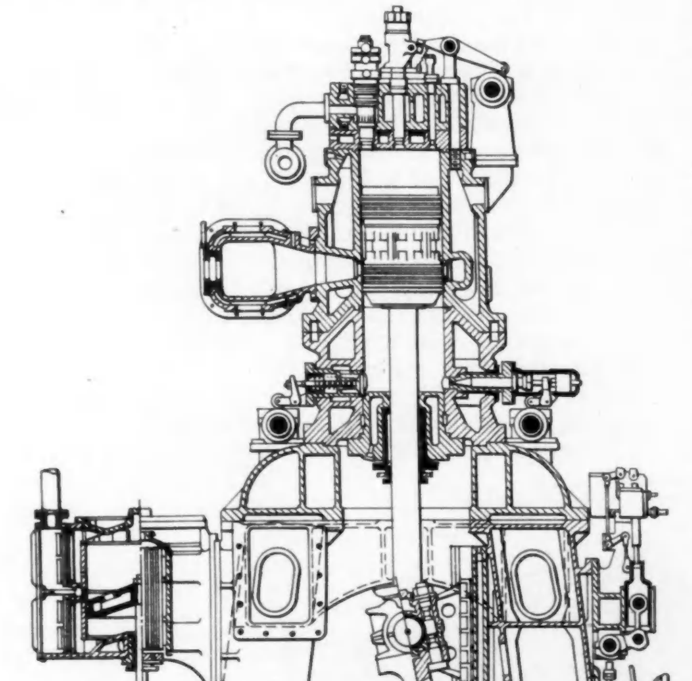


Fig. 25—A Three-Cylinder Double-Acting Two-Cycle 850-B. Hp. Engine

### Mechanical Injection and Double-Acting Diesels

To return to a marine engine of more conventional design, the Vickers engine shown in Fig. 24 is noteworthy for being the first high-pressure engine to have the airless injection of fuel that is known as solid or mechanical injection, in which the fuel is sprayed by a pump at a pressure of about 4000 lb. per sq. in. The engine shown is of the four-cycle type and develops 1250 b. hp. at 115 r. p. m. The cylinders are cast separately; they have deep bases bolted together that form an entablature which is mounted on A-type cast-iron frames that straddle the main bearings. This company also is developing a double-acting Diesel-engine, as well as a new type of opposed-piston engine, but its details are not available.

Fig. 25 is an illustration of a section of the three-cylinder Blohm & Voss 850-b. hp. double-acting two-cycle Diesel engine of the motorship Assyrian, formerly named the Fritz. The cylinder bore is 17 23/32 in., the stroke is 27 31/32 in. and the engine speed is 120 r. p. m. The stuffing-box is very deep and contains 11 sections of metallic packing; a grooved ring divides them to facilitate the distribution of lubricating oil. Each section consists of two rings split in halves that fit closely around the rod without springs, but they are kept close to the rod by a spring clip ring that is carried in a retaining ring clear of the rod. These 11 sections are entered into the stuffing-box and held in position by the gland ring. This feature was very successful and was designed in its final

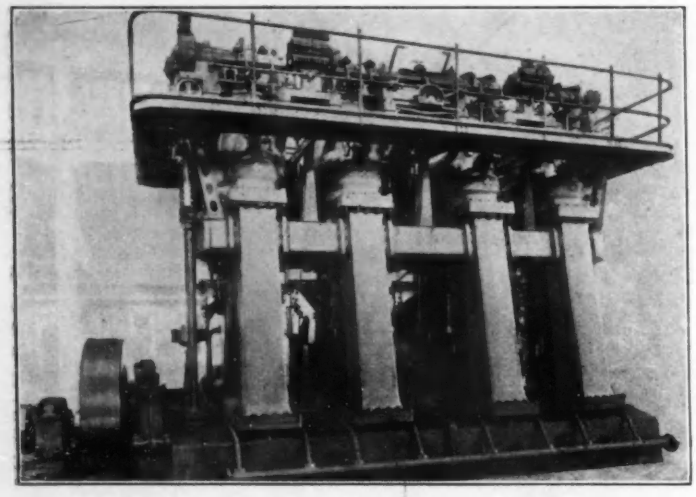


FIG. 27—A SWEDISH 1600-HP. TWO-CYCLE ENGINE

form only after months of experimental work. A prominent German Diesel engineer informed me recently that the engine itself never would be built again in its present form. The engine works without apparent mechanical difficulties, but it leaves much to be desired as to its supremacy over other systems. Its construction is of exceptionally high quality.

### COMPOUND ENGINES

The Sperry compound oil engine shown in Fig. 26, differs materially from all other designs I have illustrated. The sturdy construction necessary for an engine of its speed is indicated by the size of the crankshaft, which is considerably greater in diameter than that of any other combustion engine of which I have any knowledge and approaches the bore of the combustion cylinders themselves. The large clearance dome, which forms the combustion-chamber of the compound, stands out in marked contrast to standard Diesel practice. This dome is large and forms an upward extension of the combustion cylinder, extending also to the right in a large sweep and surrounding the transfer valve that seals the transfer port. The sleeve-like induction valve is seated on top of the transfer valve and is controlled by the cam-operated fork. The first-stage annular compression-pump, surrounding the trunk-piston below the low-pressure piston proper, delivers its air to a small receiver which, in turn, discharges to the cored port surrounding the induction sleeve. The small balancing cylinder maintains a permanent connection with the low-pressure cylinder. The solid-injection fuel-valve and nozzle are placed approximately over the center of gravity of the large masses of air in the clearance dome.

### LATEST EUROPEAN PRACTICE

Fig. 27 represents a 1600 shaft-hp. two-cycle engine that was built recently by Louis Nobel in his new factory near Stockholm, Sweden. Nobel has built hundreds of Diesel engines in 72 different designs for large tugs, tankers, gunboats and submarines on the rivers and inland seas in Russia. His first engines were installed in the tanker Sarmat in 1904 and are still in operation. The Polar engine company constructed a pair of Diesel engines from Nobel's designs for the tanker Wandal in 1903. The Sarmat's engines each developed 180 b. hp. at 240 r. p. m. and were installed in conjunction with an electric drive. With the exception of a French barge, these were the first heavy-oil-engined vessels to be built; hence Nobel has had a longer Diesel-engine experience than any other designer. Nearly all of Nobel's engines have been of the four-cycle type, but his newest creation is a two-cycle four-cylinder model of the open-crankpit type, having its cylinders cast separately and carried on heavy cast-iron A-frames. It is a very well designed piece of engineering, and compares favorably in regard to space and weight with any other design. Its fuel-consumption is 0.395 lb. per b. hp-hr. Its weight is 170 tons or 236 lb. per b. hp. The mechanical efficiency is 81 per cent and the normal operating speed is 105 r. p. m.

Another of the latest European engines is the Krupp 1400 shaft-hp. marine Diesel shown in Fig. 28. Previous Krupp merchant-marine engines were of the two-cycle type, but this new engine operates on the four-stroke cycle. It has the cylinder-beam or box construction now becoming so popular among marine designers. This is a practice that probably can be followed with advantage,

FIG. 28—A GERMAN 1400-HP. FOUR-CYCLE ENGINE

depending upon individual constructional facilities. However, it is not so suitable with two-cycle engines, because of the exhaust ports. The cylinder box in the Krupp design is carried on cast-iron columns of substantial section. The engine has many splendid engineering features.

I have endeavored to give an idea of the great variation of design that exists among the prominent Diesel engines now on the market. There are a number of others, among which are the Bethlehem-West, Craig, North British, Schneider, Ingersoll-Rand-Price, the new Worthington airless-injection engine, Sabathé, Normand, Goldberg, Steinbecker, Guldner, Atlas-Imperial, Western, Augsburg, Nurnberg, Holeby, Hocke-Fenchelle, Fletcher-Barnard, Allison, Frichs, Speedway, Washington, Knudson, Mianus-Leissner, Dodge-Hvid, Weyland, Renault, Brons-Hvid, Körting, Straus, Lübeck, Kind, Gardner, Wolverine, Kolomna and Wigelius Diesel-engines and the interesting Sumner surface-ignition type crosshead engine. All of these 37 types have their own individual outstanding features, but the 20 odd engines that I have illustrated afford a fair representation of the vagaries of modern marine-engineering progress.

### Diesel-Engine Fuel-Consumption

I wish to correct a misunderstanding that seems to exist regarding the fuel-consumption of modern Diesel-

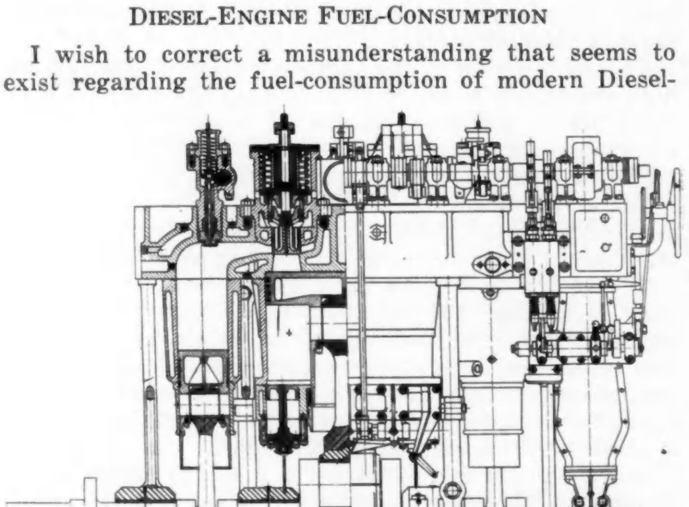


Fig. 29—Working Drawing Partly in Section of the Compound Engine Shown in Fig. 26

engined motorships. The consumption is *not* 0.50 lb. per hp-hr.; it is much less. I have been aboard many ocean-going motorships; each time I have made a special point of studying the engine-room log-book. In no instance have I ever seen recorded a fuel-consumption higher than 0.33 lb. per i.hp-hr., which was inclusive of all auxiliaries. At the most, this would be equivalent to 0.42 lb. per b.hp-hr. Generally, the fuel-consumption is somewhat lower than this, and it becomes lower after a year's operation. Sometimes, the fuel consumption is lower than 0.30 lb. per i.hp-hr. The Shipping Board's motorship William Penn averaged 0.29 lb. per i.hp-hr. on her voyage round the world.

## THE DISCUSSION

Harman Scharnagel:—During this discussion it will be of interest to explain the operation of the Sperry compound Diesel engine. The compound principle as applied in this engine is an attempt to produce a light and compact internal-combustion engine using a wide range of

Fig. 30—View of the Engine Illustrated in Fig. 26 Dismantled, Showing the Two High-Pressure Pistons, the Low-Pressure Piston and the Trunk Extension

fuels with ignition by the heat of the compression. The arrangement of the engine consists of two high-pressure four-cycle cylinders and a simple low-pressure cylinder. The high-pressure pistons are of plain trunk type. The low-pressure piston has an extension of smaller diameter

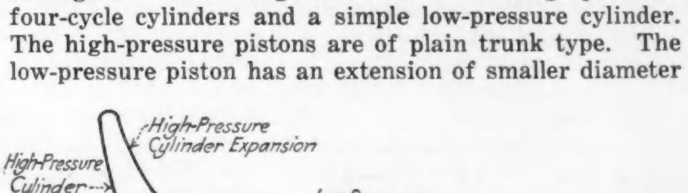



Fig. 31—Typical Indicator-Card

than the main piston. The annular space between this extension and the main piston serves as an air pump.

With reference to Fig. 28, the operation of the engine is as follows: The pump compresses air from atmospheric to a moderate pressure into a small receiver. On the down or inlet stroke of the high-pressure pistons, air under pressure from the receiver passes through the inlet-valve sleeve cooling the latter, until the pistons are at the end of the stroke. The air is then compressed on the up-stroke to about 500 lb. per sq. in. when fuel is injected. The resulting combustion and expansion of

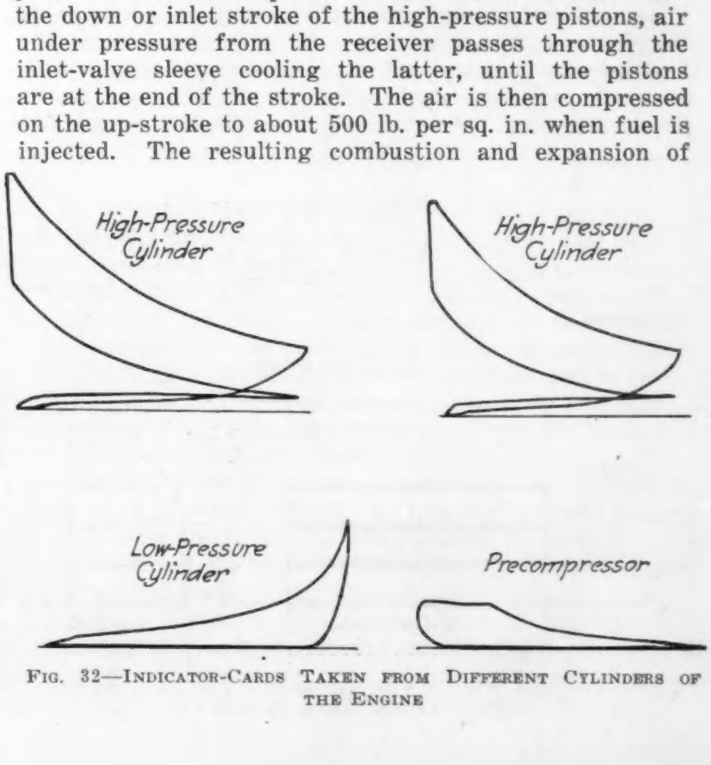



Fig. 32—Indicator-Cards Taken from Different Cylinders of the Engine

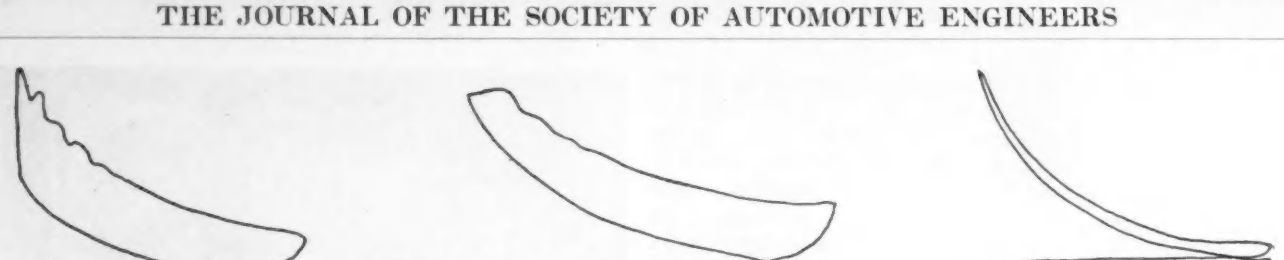


FIG. 33—CARDS OBTAINED UNDER VARIOUS OPERATING CONDITIONS
At the Left Is the Point-Top Card Obtained in the Regular Running of the Engine. By Adjusting the Fuel Injection System the Flat-Top Card in the Center Was Secured, and at the Right Is a Reproduction of a Card Taken from the High-Pressure Cylinder without Fuel

the gases drive down the high-pressure piston to the end of its stroke when the low-pressure piston, which is on the beginning of its working stroke, receives the gases from the high-pressure cylinder through the transfer port that has been opened by lifting the transfer valve from its seat into a water-jacketed cavity so that only its lower surface is washed by the passing gases. In the

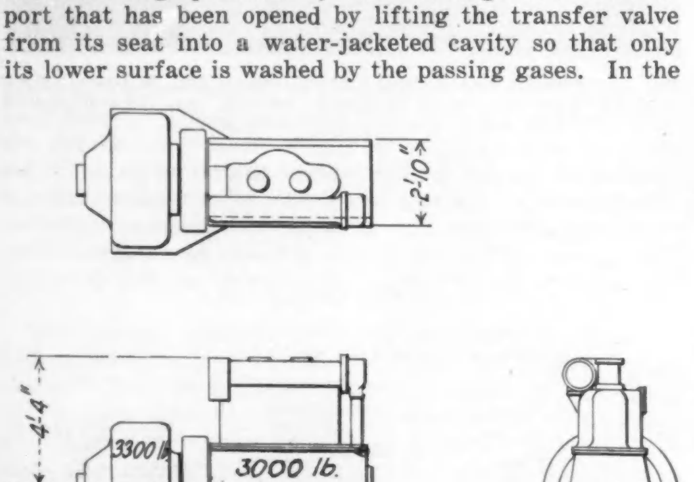



FIG. 34—THE ENGINE COUPLED TO A 50-KW. GENERATOR

low-pressure cylinder practically complete expansion of the working fluid takes place until the opening of the final exhaust near the bottom low-pressure center. Simultaneously with the working stroke of the low-pressure piston the pump space on its under side has compressed

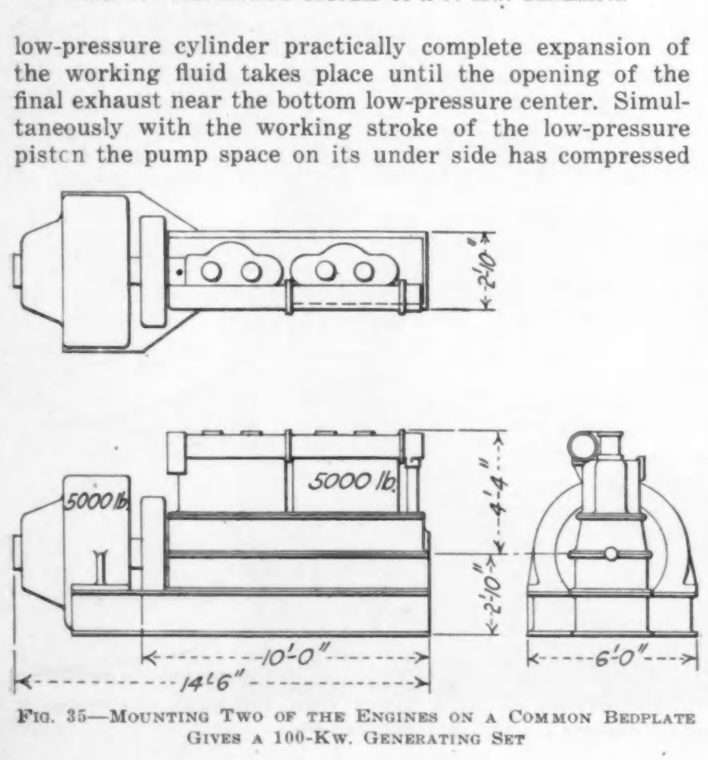



FIG. 35—MOUNTING TWO OF THE ENGINES ON A COMMON BEDPLATE GIVES A 100-KW. GENERATING SET

air into the receiver and the cycle is again carried out in the manner described. To prevent any serious drop in pressure between the high and low-pressure cylinders when the transfer takes place, the exhaust-valve is closed somewhat before the low-pressure top center and the gases are cushioned to a pressure equal to that being transferred from the high-pressure cylinder. The cranks of the two high-pressure cylinders are set together and 180 deg. from the low-pressure crank. The high-pressure cylinders fire and transfer alternately into the low-pressure cylinder so that every down stroke of that cylinder is a working stroke.

Fig. 30 is a view of the engine dismantled showing the high-pressure pistons, and the low-pressure piston with its trunk extension. The row of valves shown in the foreground are the pump inlet-valves. The delivery valves are on the opposite side in back of the piston.

The arrangement of the mechanical parts to carry out this compound cycle are such that many decided advantages are gained over the simple engine of the Diesel type. In the first place the high-pressure part of the cycle is carried out in small high-pressure cylinders which results in a great saving in weight and freedom from excessive metal heating and piston trouble. The low-pressure part of the cycle is likewise carried out in a cylinder designed for its work with advantages similar to those of the high-pressure cylinder. The same low-pressure cylinder and piston serving also as an air pump results in a twofold utilization of the same metal which means a decided saving in weight.

Fig. 31 is a diagram of the indicator-cards showing the functions of the engine as described above. The upper card shows how the regular Diesel card is divided up into the compound card. The portion of the card to the left of the vertical line is the part of the cycle carried out in the high-pressure cylinder, while that to the right is that of the pump and the low-pressure cylinder. The dotted line shows how complete expansion of the gases is obtained in the low-pressure cylinder. Analysis will show that the engine is both supercharging in the high-pressure cylinder and super-expanding in the low-pressure cylinder, thus making for economy and lightness. The lower cards show what the diagrams taken from the different cylinders look like.

The diagrams in Fig. 32 are actual cards taken from the engine. As stated before the high-pressure cylinders start on the compression stroke under a predetermined pressure from the receiver so that a much larger combustion space is needed for the volume of air after compression. This makes possible a simple means of direct fuel injection, without the use of a multi-stage air-compressor, with the resulting saving in weight and parts.

The view at the left of Fig. 33 shows the point-top card obtained in regular running of the engine due to the above combination. In the center is a flat-top card obtained by the same injection system with different adjust-

ment. At the right is a card taken from a high-pressure cylinder without fuel. The horizontal line above atmosphere shows the effect of supercharging on intake stroke with compressed air from the pump receiver.

The advantages of the compound engine are a saving in weight and floor space of between one-half and one-third in the former and more than one-half in the latter, as compared with the simple engine. The number of parts are fewer and simpler and therefore free from excessive manufacturing costs. Greater economy results on account of the complete use of the expansive force of the gases.

Fig. 34 gives an idea of what the compound engine looks like when worked up into our standard 50-kw. generating set. It is a very compact unit, its weight being about one-third that of a simple Diesel set of the same rating. By mounting two of the 50-kw. engine units on a common bedplate, we have the standard 100-kw. generating set as illustrated in Fig. 35.

Fig. 36 will interest the marine engine builders who prefer to use a reduction-gear. There are two double-unit engines connected to the reduction-gear by electromagnetic clutches giving a flexible smooth drive to the gears. This unit is laid out for 1800 shaft hp., the engine running at 250 r.p.m. with a tail shaft speed of about 90 r.p.m.

Fig. 37 shows the comparison between the compound engine and a simple Diesel engine for direct drive with the same piston speed and revolutions.

The engine shown averages about 0.5 lb. per b. hp-hr.

J. D. GILL:—What is a Diesel engine?

T. O. LISLE:—There are two essential elements of characterization in the true Diesel cycle that are not found in any other class of engine. There is also a third essential feature, but it is not so rigidly confirmed in its application. The three fundamental features are

(1) Compression of pure atmosphere to a degree such that the temperature produced is adequate for the inflammation and combustion of the fuel
(2) Injection of fuel at such a rate that the burning proceeds without a rise of pressure in the combustion space. This condition is not realized with absolute precision, there being always a slight rise of pressure when the fuel begins to burn
(3) Injection of fuel by air-blast that produces the turbulence needed for good combustion. This is essential, but it is not distinctive of or exclusive in the Diesel cycle

An engine either has these features or it does not have them; correspondingly, it is of the Diesel type or it is not. No air-blast was used for the first experimental engine that was built by Dr. Diesel. It was only after air-injection had been employed, when the first engine was rebuilt, that a sufficient number of revolutions per minute were obtained to secure indicator-cards of the whole cycle. The initial combustion obtained from the first engine blew the indicator to pieces. The true Diesel cycle, as now in worldwide use, is not in accordance with the master patents taken out in 1892, before Diesel made his conclusive experiments. It is in accordance with the design of the second engine, which was put on the test-bed at Augsburg in 1896. The final experiments that led to success were made upon this second engine, and 4 years were devoted to them. This engine used air-injection; hence, air-injection of fuel is an essential feature of the Diesel cycle. But this is not a distinctive or exclusive feature, because air-injection of fuel is used with oil engines that do not contain the other essential features of the Diesel cycle.

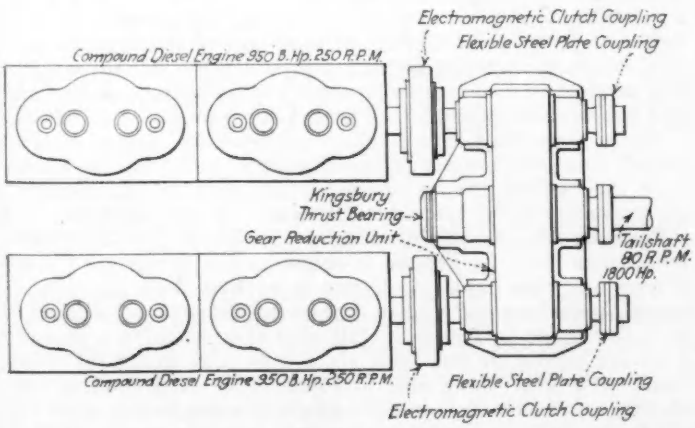



FIG. 36—TWO DOUBLE-UNIT ENGINES ARRANGED FOR MARINE USE BY CONNECTION TO A REDUCTION GEAR THROUGH ELECTROMAGNETIC CLUTCHES

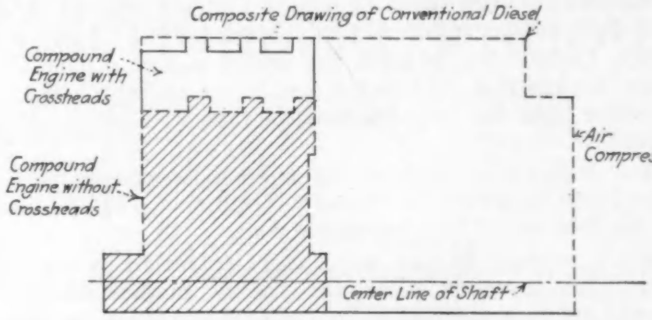



FIG. 37—COMPARISON BETWEEN THE COMPOUND ENGINE AND A SIMPLE DIESEL ENGINE FOR DIRECT DRIVE AT THE SAME PISTON SPEED AND NUMBER OF REVOLUTIONS

MR. GILL:—I asked for a definition of what a Diesel engine is because we have many engines today that are not Diesel engines according to the original conception and yet, if we include them in the list that has been given, they are Diesel engines. I think that we must draw the line somewhere. The Diesel cycle is what is called a constant-pressure cycle and, in the automobile engine, we have the constant-volume cycle. Dr. Diesel succeeded in obtaining pressures as high as 300 lb. per sq. in., but he did not have suitable material to withstand extremely high pressure. Nowadays, engines work at a pressure of 300 to 500 lb. per sq. in.; with solid injection, this means that there is no air condensation. Some of the Sperry class of engines hold to the same lines.

J. W. NORTON:—We can hardly say that the semi-Diesel engine holds to the same lines or class. The compression pressure is smaller, and some means of retaining the heat for the next charge is provided, such as the hot plate and hot bulb. These engines work around 300-lb. compression without spark-plugs, whereas the pure Diesel engine works around 400 to 500 lb. per sq. in. The diagram of a Diesel engine working on a constant-pressure cycle resembles that of a steam engine with a 10 to 12 per cent cut-off. The semi-Diesel and the automobile engine are mainly of the explosion or constant-volume cycle type and the diagram reaches a sharp peak, with the termination of the compression pressure half-way up, while in the true Diesel engine the compression line runs all the way up to the admission line.

CHAIRMAN HERMAN HOLLERITH, JR.: — I believe the fundamental difference to be that the Diesel engine burns its fuel at a constant pressure, and that the automobile engine burns its fuel at a constant volume. There are

variations between them. I believe that the engines Mr. Lisle has mentioned are all full Diesel engines.

MR. LISLE:—No, because the letters patent distinctly state that the fuel shall be injected by air and at a higher pressure than the compression pressure within the cylinder.

MR. FRENCH:—None of the engines mentioned is a true Diesel engine. Dr. Diesel's invention called for an engine that would burn powdered coal.

MR. NORTON:—It was Dr. Diesel's intention to use a very high pressure after the theory mathematically worked out that the higher the compression pressure, the higher the thermal efficiency. In the Diesel engine *pure air* is compressed and fuel is injected at the highest pressure. That is why the Diesel engine can work with high pressure-ranges. In engines using gasoline and kerosene, or in other words, volatile fuel, the pressure range is lower, due to a premixture of air and fuel, and possible backfire. Dr. Diesel used coal as fuel in his first engine and a compression-pressure around 3500 lb. per sq. in. It did not work out according to expectations, so it was changed to burn fuel oil and the compression-pressure lowered.

The two-cycle and four-cycle engines that Mr. Lisle has shown can be made both single and double-acting, but they are the only types we have. The difference in appearance of the engines is due solely to the mechanical features of the two types.

JOHN H. BARNARD:—I am still working with my engine to obtain regularity of performance. Like many experimenters, I began with too small an engine. It has only an 8-in. bore and a 10-in. stroke. That calls for only 0.01 cu. in. of fuel oil per stroke. Furthermore, it requires that the oil be sprayed progressively; in other words, the amount of oil delivered into the combustion-chamber must increase as the volume above the piston increases. Hence, instead of having the problem of trying to inject a small quantity of oil at what might be called an even rate of injection through a comparatively few degrees of the crank angle, which in the Diesel type averages 15 deg., I am trying to spray oil progressively through about 64 to 70 deg. of the piston stroke.

These spray requirements constituted the difficult part of the problem, but that has been solved. The present difficulty lies in the regularity of performance. It is difficult to obtain regularity with so small a fuel pump. Ordinarily a pump is used with Diesel engines that has a capacity considerably in excess of the engine's requirements with means for by-passing so much of its delivery as is in excess of the amount to be metered for each stroke. However, I cannot do that with progressive spraying, and it is very difficult to produce a small pump that will throw 0.01 cu. in. of fuel with absolute regularity. I have been forced into using an entirely novel form of pump. It is just now being made and I hope it will solve the difficulty, because this has now been located absolutely in the irregular performance of the fuel pump.

CHAIRMAN HOLLERITH:—Mr. Barnard has expressed one of the prime limitations of the small-sized Diesel engine; that is, the proper measuring of such very small quantities of fuel.

HARTE COOKE:—The fundamental difference between the Diesel cycle and other cycles is that, in the Diesel engine, only the air is compressed and, in the others, the explosive mixture is compressed to obtain the high efficiency required. With the ordinary engine the compression-ratio is limited if the explosive mixture will preignite. The ratio can be made whatever one pleases. Dr. Diesel originally intended to reduce the fuel at such a rate that it would burn at a constant temperature; in other words, he burned the fuel as the heat was extracted by the work on the piston. He found that this was not feasible; so, he used the constant-pressure cycle.

# MUTUAL BENEFITS OF FOREIGN TRADE

(*Concluded from page 90*)

people is in their own powers of production. They must be able to use those powers and to pay in the products of their own labor or they cannot buy.

We are much inclined to magnify the danger of competition from other countries on equal terms. We hear much about the ability of this or that country to overrun and dominate the markets of the world. We used to hear that England could do it, but of late years Germany has been the bugaboo. It is possible, of course, that in given lines of industry certain countries may be superior to others; that situation is the very basis of international trade. There is not the least danger, however, that of its own choice any country will do more than its share of the work of the world. No people can work any more than all the time, and none of them want to give their goods away. They all want to get something for them. The great exporting nations always have been great importing nations. They have to be. They cannot sell their own products without taking the products of other countries. We have seen how trade balances in our favor have caused the exchange rates to rise against us, and put a check upon our exports.

The great lesson of the time is that of *the mutual interests which, rightly understood, tend to unite all countries in reciprocal and helpful relations.* Unfortunately, there is only a faint comprehension of the truth; and because this is so we have a world of rivalries and antagonisms which from time to time break out in war. The leading business interests of every country have a responsibility in this respect. The spirit of war is developed in mistaken ideas about national interest. If nations believe that their fundamental interests are in conflict, that there is irreconcilable rivalry and a struggle for existence, if people believe that the future of their country and their children is at stake, of course they will fight; nothing else can be expected.

But there is no such conflicting interest. There are trade rivalries, rivalries between traders of different countries, as there are between traders of the same country. Within proper limits they are stimulating and beneficial. But it is a mistake to emphasize them as though the success of one nation depends upon driving another out of the field. That idea is based upon the assumption that there is only a limited amount of business to be done and never enough for all and has been responsible for an infinite amount of mischief.

There is no limit to the amount of work to be done in the world, or of the amount of business to be had, because there is no limit to the amount of wealth that may be created from the resources of nature, or to the wants of the population.

We hear considerable about over-production, but there can be no such thing as general over-production of all the goods of common consumption. There is such a thing as *unbalanced* production, and we see it now; but there is not a family in a four-room tenement in this city that would not like to have six rooms, or one with six rooms that would not like eight, with everything else to correspond. The problem of society is to organize and integrate the productive powers of all countries to secure the greatest possible supply of the comforts of life for the population. That is the great appeal to the constructive forces of the world.

# Tractor and Plow Reactions to Various Hitches

*By* O. B. Zimmerman[1] and T. G. Sewall[2]

Minneapolis Tractor Meeting Paper *Illustrated with* Photographs and Charts

The authors enumerate some of the questions that are involved and, after outlining a previous paper on the subject of plows, analyze these questions in part by the aid of diagrams and applied mathematics. Comparative draft data are presented in tabular form and commented upon, as well as comparative hitch-length data.

Tractor reactions are explained and discussed in some detail in a similar manner, special attention being given to the reactions on a slope and up-hill. The reactions on cross-furrow slopes are considered, comparisons being made between two tractors that were reported upon in the University of Nebraska tests. The factors involving tractor stability and resistance against overturn are analyzed. The authors state that the analysis presents a definite method of attack for the more correct solution of the proper hitching-point, as well as being a study relating to lug design.

Those familiar with the application of farm tractors to field use are well aware of the wide range of arguments concerning the questions of how to hitch the implements to a tractor and whether to run the tractor on the unplowed land or with one wheel in the furrow. Other questions are whether to hitch the plow low or high on the tractor; whether the point of hitch shall be to the right or left of the plow center; where the center of plow reaction is located; what the nature of the reactions is; what the changes are in the several reactions during motion as regards weight and stability; and how these changes affect tractor stability against overturn. The paper endeavors in part to analyze the answers to these questions.

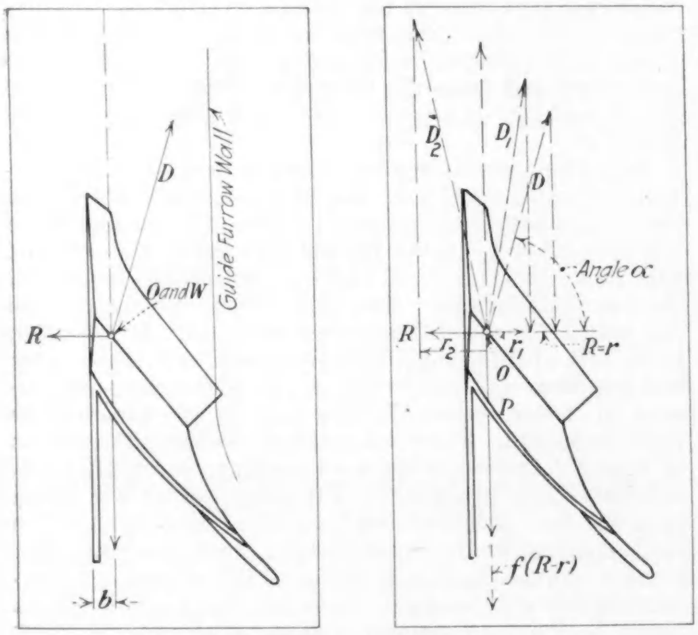



Fig. 1—Reactions with an Individual Plow

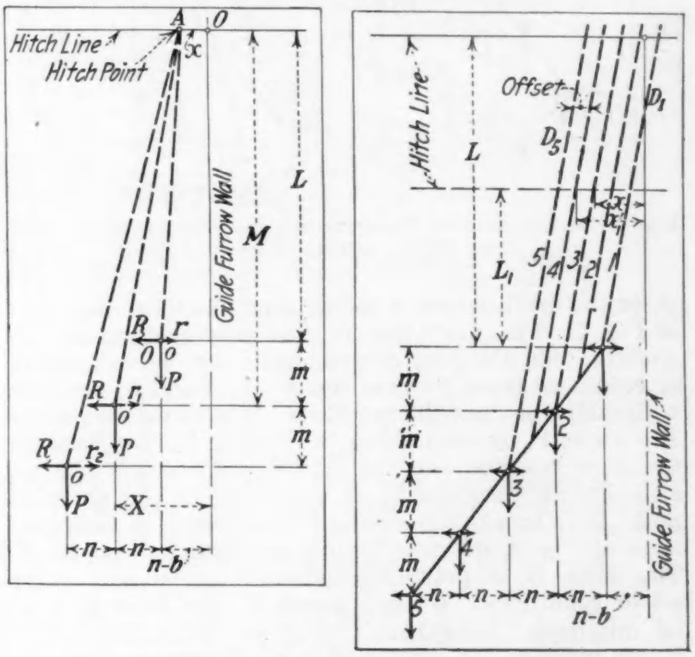



Fig. 2—Diagram Showing the Necessity for Exercising Care in Connection with the Hitch of a Multiple Plow

In an article on Coordination of Theory and Practice in the Design and Operation of Plows, by A. C. Lindgren and O. B. Zimmerman,[3] which covers the subject of plow reactions, a careful analysis is given as regards the plow and this subject can be reviewed in its extended form in that article. But it is now desirable to carry this analysis still farther so as to cover the tractor actions and reactions also, and so make it more complete. Therefore, a review of the deductions of the article to which reference has been made becomes necessary.

In brief, we must start with the individual plow shown at the left of Fig. 1, in which the complex series of primary reactions can be considered best as having been resolved and integrated into one vertical force, $W$, one cross-furrow force, $R$, and one down-furrow force, $P$. These reactions will vary in magnitude as they are developed, due to the weight of the outfit, the soil operated upon, the condition of the plow, the shape of the moldboard, the speed of operation, the depth of plowing, the width of cut, the nature of the soil, the moisture content, the relative friction between steel and earth, and the like. It can be shown how these forces may be considered as acting upon one point, $O$, which will be called the center of reaction of the plow.

To overcome these reactions, we have the active force,

[1] M.S.A.E.—Engineering staff, International Harvester Co., Chicago.
[2] Experimental department, International Harvester Co., Chicago.
[3] See *Journal of the American Society of Agricultural Engineers*, January 1922, p. 3.

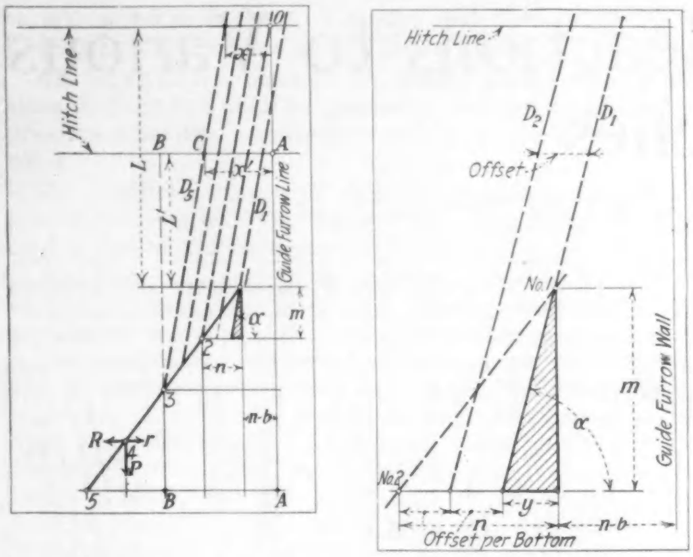



FIG. 3—DEVELOPMENT OF FORMULAS FOR THE CORRECT HITCH WITH MULTIPLE RIGID UNITS

$D$, or the draft, which is shown more clearly at the right of Fig. 1. This draft line, $D$, may vary in direction. A study of this direction of pull shows that we can add to or reduce all three primary forces, $W$, $R$ and $P$, but very noticeably the cross-furrow force, $R$, and the down-furrow force, $P$, by overcoming the sliding friction between the plow landside and the furrow wall, which friction causes wide variation in $D$, according to the angle of pull. If $D$ acts to the right of $O$ as shown in both portions of Fig. 1, its cross-furrow component $r$ opposes $R$. The force $R$ is practically constant under any given set of conditions. When $r$ equals $R$, the force $D$ is at its minimum. Likewise, if we direct $D$ to the left of $O$, we increase the total to $R + r$. However, a reasonable pressure in the direction of the furrow wall is necessary for stable action. The argument concerning this feature should be reviewed in the article that has been cited.

The multiple-plow unit quickly shows us the necessity of care in the matter of hitch because, as shown in the left half of Fig. 2, if each individual plow is permitted to receive its draft $D$ from a common hitch-point, it is clear that the cross-furrow action of $R$ and $r$ must be different; hence, we must unite them rigidly or provide at least for possible interchanges of pressure between them so that they will work as a unit. Otherwise, each plow will tend to interfere with the others. Plow No. 1 next the furrow wall tends to crowd to the left, and plow No. 3 crowds to the right. The right-hand portion of Fig. 2 and both drawings in Fig. 3 illustrate the development of the formulas for the correct hitch in multiple rigid units.

Let

$\alpha$ = Angle of $D$ with respect to the cross-furrow line
$b$ = Distance of center of individual plow reaction to plow landside
$D$ = Draft in the drawbar, to obtain the relations of draft
$f$ = Coefficient of friction of earth on steel
$L$ = Distance of hitch line from the center of reaction of plow No. 1
$M$ = Spacing between plows, down furrow
$N$ = Number of plow bottoms
$n$ = Width of cut of a single plow
$O$ = { Location of combined center of reaction from the furrow wall; Hitch distance from the hitch line to the center of reaction }
$P$ = Down-furrow draft without furrow-wall friction
$R$ = Cross-furrow reaction due to shape of moldboard, etc.
$r$ = Cross-furrow component of $D$
$x$ = Hitch-point distance from the furrow-wall measured on the hitch line

Then

$$D = \sqrt{([P + f(R - r)]^2 + r^2)} \quad (1)$$
$$O = (w - b) + (N - 1)\, n/2 \quad (2)$$
$$O = L + (N - 1)\, m/2 \quad (3)$$
$$x = (n - b) + [N - 1] \times ([(n - m) \cot \alpha] \div 2) - L \cot \alpha \quad (4)$$
$$\tan \alpha = \frac{P + f\,(R - r)^2}{r} \quad (5)$$

The conclusion follows that the correct hitch direction should lead to the right of $O$ in all cases by an amount that will reduce the friction to a practical minimum. Here is where the tractor plays its part and receives these reactions through the various styles of hitch, either rigid or flexible. The basic principle of flexible-hitch forms is illustrated in Figs. 4 and 5. From the foregoing text and from the illustrations, we can determine the desirability of long or short hitches, and high or low hitching.

In the field, we have a considerable range of hitch distance from the furrow wall, any point of which will answer for practical operation. Beyond these limits, we find that if we hitch the tractor drawbar to the right of this range the plows will run into the plowed land until the reactions balance. The plows then will not cut the full width for which they were designed. If we hitch to the left of this range, the furrow wall will crush down and the plows will run to the left until the reactions balance in that direction, leaving part of the area at the right unplowed. Standard tractor-drawbar adjustments in regard to plow-hitches were adopted March 11, 1922, and have been published.[4] The values given are based upon the best practical and theoretical arguments. The correct limits to the right are for light or fluffy soil; hence, as shown in Fig. 5, various hitch points are indicated from light, medium, heavy and very heavy soil as shown by draft values and locations $L$, $M$, $H$ and $V\ H$. These values indicate that for an equal cross-furrow reaction, ($R$-$r$), the heavier soils require a hitch point closer to the furrow wall than lighter soils.

[4] See S. A. E. HANDBOOK, p. K-40.

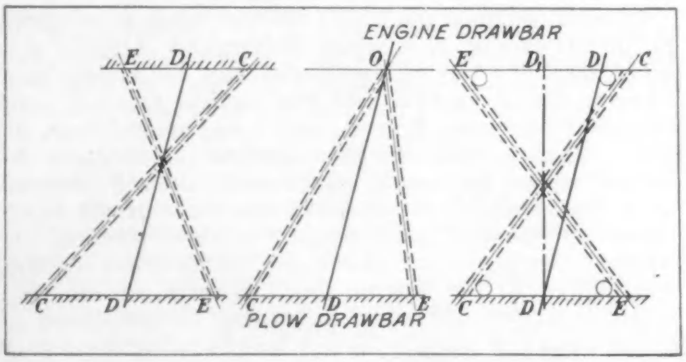



FIG. 4—DIAGRAM ILLUSTRATING THE BASIC PRINCIPLE OF THE RIGID HITCH FOR PLOWS

## COMPARATIVE DRAFT DATA

Figs. 6 and 7 are given with Tables 1 to 6 to emphasize the various relations that result from varying the hitch point under a given set of data, assuming that the plow units are designed without complete relief to the cross-

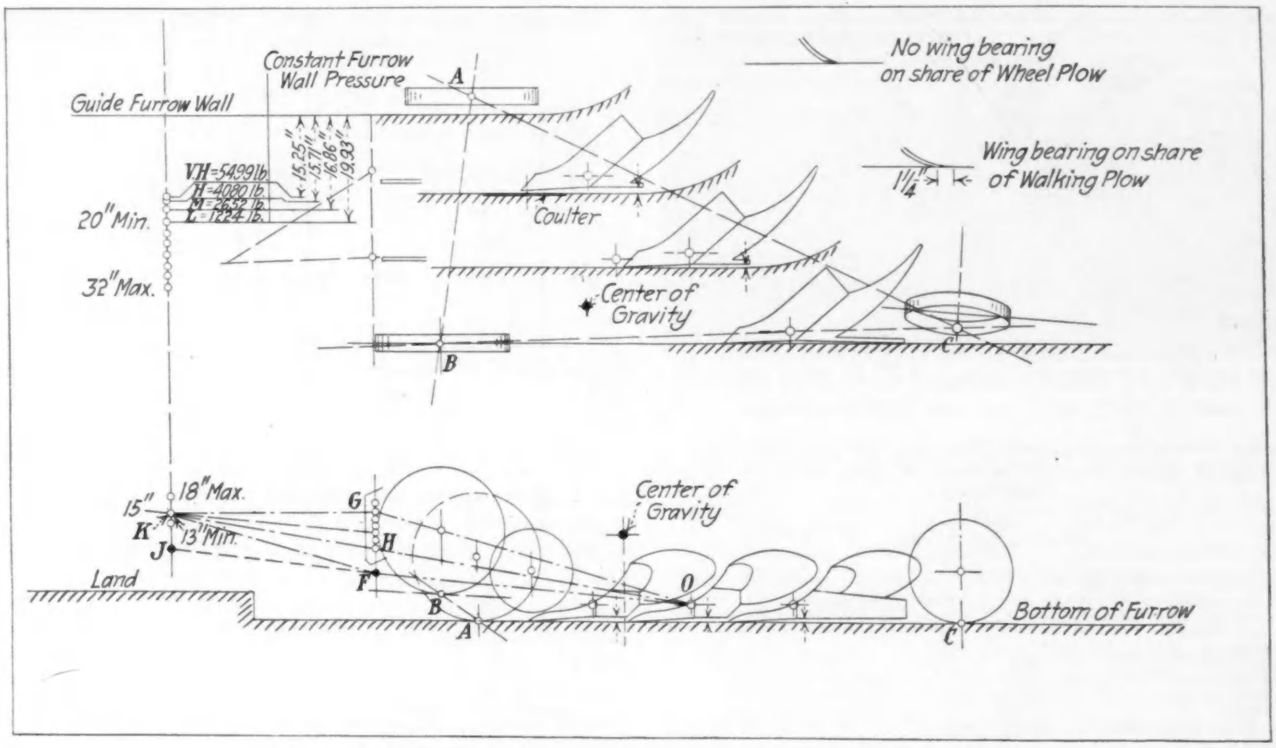



FIG. 5—HITCH POINTS FOR LIGHT, MEDIUM, HEAVY AND VERY HEAVY SOILS

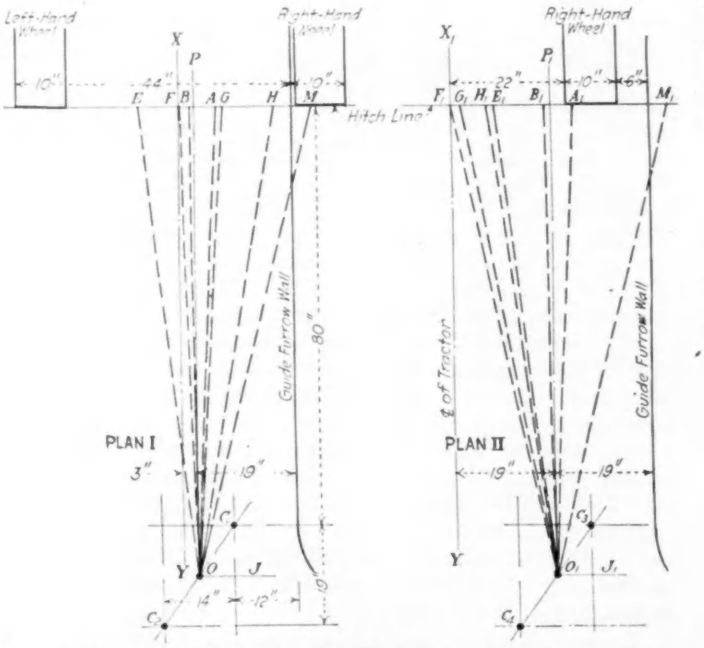



FIG. 6—RESULTS OF VARYING THE HITCH POINT FOR A GIVEN SET OF CONDITIONS WITH A TWO-BOTTOM PLOW

furrow pressure by carrier wheels. While these carrier wheels ease off the plow pressure, we must still take note of their values since the wheel bearings or the tractor as well as the plow must take up these pressures; hence, a knowledge of their magnitude is valuable. Fig. 6 represents the conditions when a tractor pulls a plow having two 14-in. bottoms. The overall dimensions and tractor wheel widths are taken as being averages of the output of some 20 prominent manufacturers. Plan I represents the conditions when the tractor runs in the furrow; in Plan II it runs on the land. The lines $O\ P$ and $O_1\ P_1$ are down-furrow lines through the center of reaction of the plow unit, and lines $X\ Y$ and $X_1\ Y_1$ are the center lines

TABLE 1 — DISTANCES FROM THE CENTER LINE OF THE TRACTOR OF POINTS OF EQUAL DRAFT; TWO BOTTOMS

| | Point Distance, In. | | | |
|---|---|---|---|---|
| Plan I | $M = +25.4$ | $A = +7$ | $B = +2$ | $E = -8$ |
| Plan II | $M_1 = +41.4$ | $A_1 = +23$ | $B_1 = +18$ | $E_1 = +8$ |

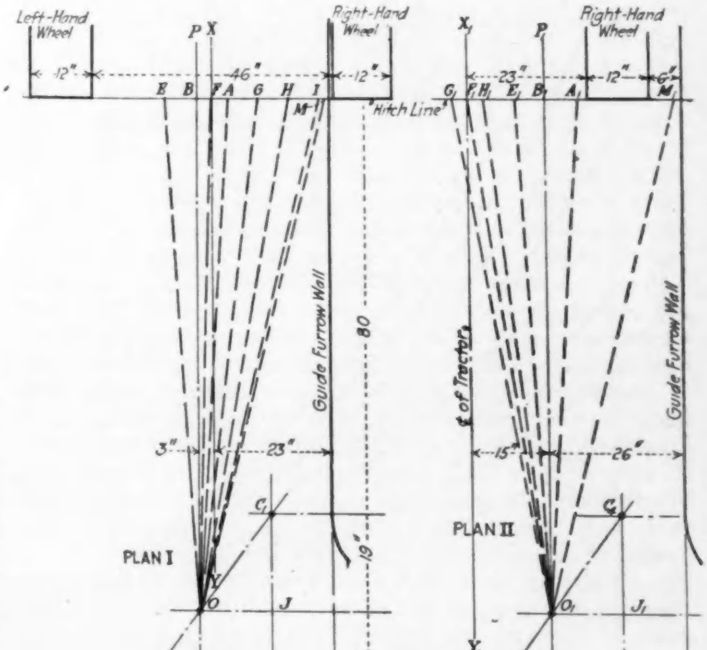



FIG. 7—RESULTS OF VARYING THE HITCH POINT FOR A GIVEN SET OF CONDITIONS WITH A THREE-BOTTOM PLOW

TABLE 2—DISTANCES FROM THE FURROW-WALL OF POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; TWO BOTTOMS

| | Point Distance, In. | | | | |
|---|---|---|---|---|---|
| Plan I | $F=+22$ | $B=+20$ | $A=+15$ | $G=+14$ | $H=+4$ |
| Plan II | $F_1=+38$ | $G_1=+36$ | $H_1=+31$ | $E_1=+30$ | $B_1=+20$ |

of the tractor. The points $A$, $B$ and $E$ and $A_1$, $B_1$ and $E_1$ represent the hitching points as recommended by the Society; namely, 15 in. as a minimum, 20 in. as a proper or best average position and 30 in. as a maximum,

TABLE 3—TOTAL DRAWBAR PULL AT POINTS THE SAME DISTANCE FROM THE CENTER LINE OF THE TRACTOR; TWO BOTTOMS

| | Drawbar Pull at Point | | | | |
|---|---|---|---|---|---|
| Plan I, lb. | $F=1,033$ | $B=1,010$ | $A=958$ | $G=949$ | $H=869$ |
| Plan II, lb. | $F_1=1,188$ | $G_1=1,170$ | $H_1=1,124$ | $E_1=1,116$ | $B_1=1,010$ |
| Increase, Plan II over Plan I, lb. | 155 | 160 | 166 | 167 | 141 |
| Increase, Plan II over Plan I, per cent | 15.0 | 15.8 | 17.3 | 17.6 | 16.2 |
| Average per cent of Increase | .......... | .......... | 16.4 | .......... | .......... |

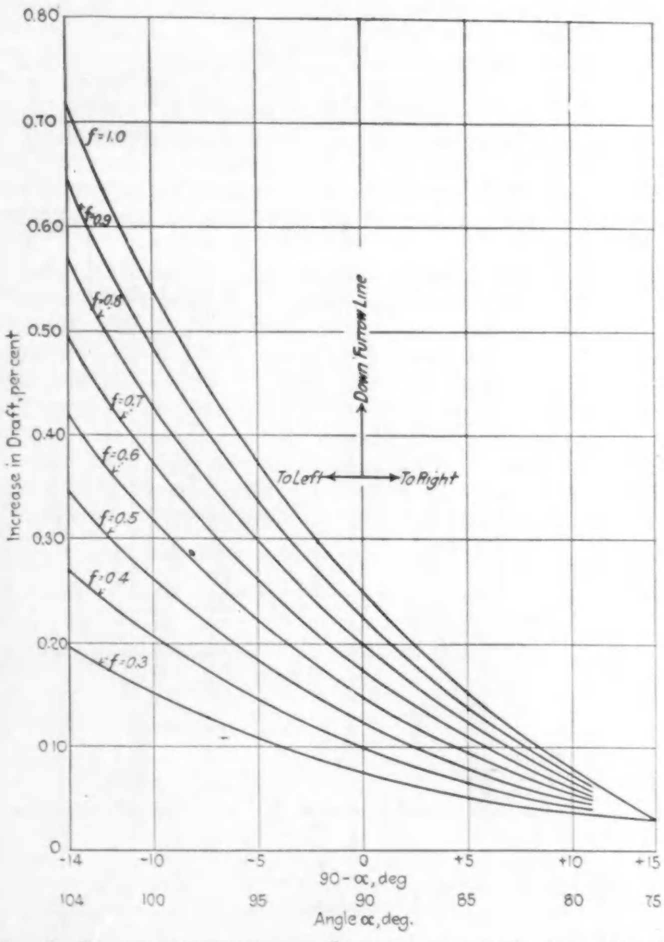



FIG. 9—RELATION BETWEEN THE INCREASE IN THE DRAFT AND THE ANGLE THAT THE LINE OF DRAFT MAKES WITH THE CROSS-FURROW LINE

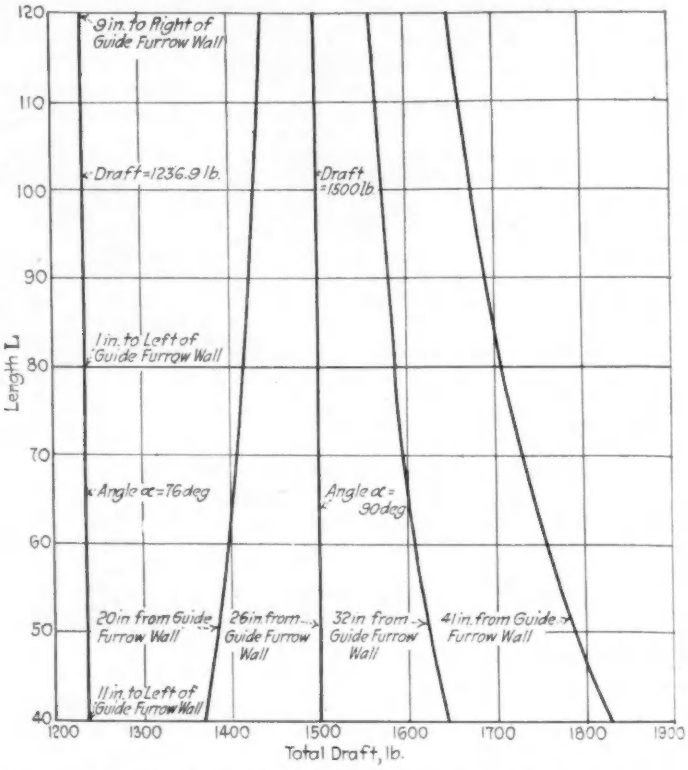



FIG. 8—RESULTS OF CHANGING THE LENGTH OF HITCH BETWEEN THE DRAWBAR OF A THREE-PLOW TRACTOR AND THE FIRST PLOW

TABLE 4—DISTANCES FROM THE CENTER LINE OF THE TRACTOR OF POINTS OF EQUAL DRAFT; THREE BOTTOMS

| | Point Distance, In. | | | |
|---|---|---|---|---|
| Plan I | $M=+21.75$ | $A=+3$ | $B=-3$ | $E=-9$ |
| Plan II | $M_1=+39.75$ | $A_1=+21$ | $B_1=+15$ | $E_1=+9$ |

respectively, from the guide-furrow wall. A comparison of the hitching points as to ($a$) distance from the center line of the tractor and distance from the guide-furrow wall and ($b$), drawbar pull, is given in Table 1, using formulas (1) and (5).

The lines $M$ $O$ and $M_1$ $O_1$, have the angle $\alpha$ equal to 76 deg., where the draft is least, on account of the removal of all side pressure against the furrow wall in the case taken. The drawbar pull at the points $A$, $B$, $E$ and $A_1$, $B_1$, $E_1$, for both plans is the same. A tractor running in the furrow, therefore, can use a hitch nearer to the center line of the tractor.

### COMPARATIVE HITCH-LENGTH DATA

Fig. 8 is developed to emphasize still further the values resulting from changing the length of hitch between the first plow and the tractor drawbar.

It is clear that if we desire the tractor to operate on the land in preference to operating in the furrow, and

TABLE 5—DISTANCES FROM THE FURROW-WALL OF POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; THREE BOTTOMS

| | Point Distance, In. | | | | | |
|---|---|---|---|---|---|---|
| Plan I | $B=+26$ | $F=+23$ | $A=+20$ | $G=+14$ | $H=+8$ | $I=+2$ |
| Plan II | $G_1=+44$ | $F_1=+41$ | $H_1=+38$ | $E_1=+32$ | $B_1=+26$ | $A_1=+20$ |

TABLE 6—TOTAL DRAWBAR PULL AT POINTS EQUI-DISTANT FROM THE CENTER LINE OF THE TRACTOR; THREE BOTTOMS

| | Drawbar Pull at Point | | | | | |
|---|---|---|---|---|---|---|
| Plan I, lb. | $B$=1,500 | $F$=1,457 | $A$=1,417 | $G$=1,348 | $H$=1,290 | $I$=1,242 |
| Plan II, lb. | $G_1$=1,746 | $F_1$=1,709 | $H_1$=1,670 | $E_1$=1,588 | $B_1$=1,500 | $A_1$=1,417 |
| Increase, Plan II over Plan I, lb. | 246 | 252 | 253 | 240 | 210 | 175 |
| Increase, Plan II over Plan I, per cent | 16.40 | 17.30 | 17.85 | 17.80 | 16.28 | 14.10 |
| Average, per cent of increase | .......... | .......... | 16.62 | .......... | .......... | .......... |

with hitches near the tractor center, a long hitch is desirable with two and three-plow units. If we develop formula (1) through all the varying angles within the range of practical operation from right to left, we begin to realize the effect that results from choosing a direction of draft $D$ correctly, especially for outfits which do not provide for relief of the $R$ or cross-furrow pressure through carrier wheels, which not only support the unit but can relieve this thrust when set at a slant to the vertical so as to oppose this thrust. The several values of $f$ are some of the coefficients of friction between various earths and steel. In Fig. 9, the 90-deg. angle represents a straight down-furrow draft through $O$, the center of plow reaction. The 85, 80 and 75-deg. angles indicate the angles of draft that $D$ makes with respect to a cross-furrow line and, likewise, the 95, 100 and 105-deg. angles represent similar draft lines to the left of $O$.

The vertical scale indicates the percentage of increase above the original value of $P$, such as $P$ = 400 lb. When $D$ is pulling so that the angle $\alpha$ = 85 deg. in soil that has a coefficient of friction of 0.5, we find that the draft is 8 per cent higher than the base figure $P$. If we should set the hitch so as to develop a pull 10 deg. to the left with $f$ = 0.5, we would have a draft 26 per cent higher than the base figure, which is that much higher than is necessary.

An attempt to visualize the series of variables is given in Fig. 10. Here we have a space relation in which the three coordinates are as follows:

(1) The ratio of $P$ to $R$ or the relation of the down-furrow force $P$ to the cross-furrow force $R$ that exists at the moment under consideration when the plow is in operation and when we are comparing the various kinds of moldboard, those having a long slope to those the slope of which is very abrupt. These ratios have been deduced only roughly from judgment and practical operation. The diagram necessarily does not cover all of them, but it serves a useful purpose

(2) The angles of pull of $D$ between the center of plow reaction and the hitch point, with respect to either a cross-furrow line like $R$ having values of $\alpha$ from 90 to 75 deg. or, as shown above, as the angle to the right of a down-furrow line like $P$, such as 90 deg. — $\alpha$. The effect of angles of pull to the left of the down-furrow line, or those more than 90 deg., can be interpreted roughly by imagining an extension of the warped planes through the left coordinate planes and in accord with Fig. 9

(3) The percentage of increase of pull $D$ over and above that needed is shown in percentages, from 0 to 25. As the draft $D$ is affected markedly by the coefficient of friction $f$, this feature is incorporated also. These relations form the surfaces that appear like the leaves of a book

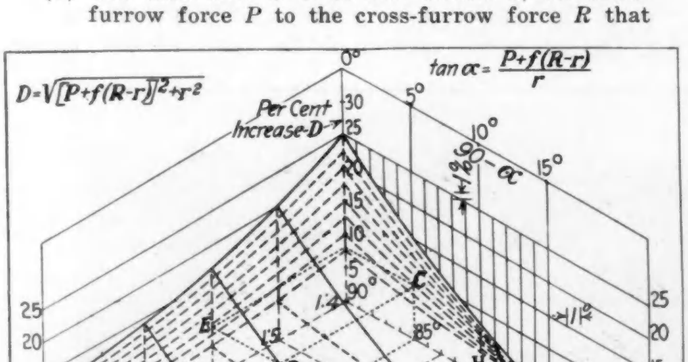



FIG. 10—THREE-DIMENSION DIAGRAM SHOWING THE INTER-RELATION OF THE SPEED, KIND OF SOIL AND TYPE OF MOLDBOARD

FIG. 11—AN 18-BOTTOM PLOW BEING DRAWN BY A SINGLE TRACTOR AT THE RATE OF 1 ACRE IN LESS THAN 12 MIN.

To interpret Fig. 10, let us consider a case where we are hitching so that the draft line $D$ makes $\alpha$ = 85 deg. or 5 deg. from the down-furrow line and pulls to the right. Run down the line marked 85 deg. to $F$, which lies in the cross plane that represents $P$ to $R$ as being 6 to 1, then rise until the line pierces the plane as at $D$ where $f$ = 1. Then $F$ $D$, read on the percentage scale, shows 8 per cent. This 8 per cent less $X$, which is 1 per cent, gives us a 7-per cent, net, greater draft for $D$ with a hitch at 85 deg. than would be shown if the hitch had been set at $H$, or at 81.5 deg. run out from $x$. At 81.5 deg., the cross-furrow force would be zero; hence, the practical angle would be somewhere between 81.5 and 85 deg., according to the desired furrow-wall pressure and such other practical accommodation as is necessary.

A review of Fig. 10 will clear up the value of these relations, when thinking relatively of light and heavy-soil operation. We can say then that when a plow is operating in a field of varying resistance, the draft $D$ will vary with a given angle of hitch, passing, say, from plane $f$ = 1 to plane $f$ = 0.6 back and forth. Also, as the speed might vary, causing the $P$ to $R$ relation to go from 1 to 6 to 1 to 5, and if for any reason the angle of hitch should vary a few degrees due to hill-slope plowing, say between 4 to 7 deg. from the down-furrow line, the draft would follow the corresponding relations according to the several coordinate values. This emphasizes the desirability of attempting to establish better the technical relations of $P$ to $R$. These relations should be developed and also the values of $f$ covering the typical soils tilled in the

United States; also, the relation existing due to the changes of $f$ for the various degrees of moisture content. These are real research problems in new plow design.

The foregoing deductions are correct for large units pulling numbers of bottoms. In Fig. 11, a plow having 18 bottoms is being drawn by one power unit and this unit is plowing at the rate of 1 acre in less than 12 min., the center of the tractor being seen to be just ahead and between plows Nos. 6 and 7, counting from the furrow wall, thus leaving 11½ plows to the left. Fig. 12 gives an example of the mass cooperation of three 45-hp. tractor units that are pulling plows having 55 bottoms, again showing the application of minimum draft by relief of the cross-furrow reaction allowing perfectly true draft. The clean-cut furrow-wall is evident at the left, with no crushing down in this light soil. The resistance was about 330 lb. per 14-in. bottom. An area was plowed with the larger unit at the rate of 1 acre in less than 4 min.

### Tractor Reactions

The development of the following formulas and discussion of them will apply equally well to any self-propelled wheeled type of motive power, carrying loads, hauling or doing both, over level ground, up and down hill or on hillsides where the slope affects side tipping. This includes tractors, trucks, automobiles, power-driven cultivators and the like.

Starting with the engine as the source of power, there is a limit to its average maximum effort. For successful continuous operation, the amount of power at the source must be greater than the sum of all resistance; also, means for exerting this power must be provided through wheel reaction to the soil underneath the tractor. Therefore, the engine power must be greater than the friction of transmission, plus the load, plus the rolling resistance, plus any lifting of the tractor due to the slope of the land traversed. In this paper we are concerned principally about the effects of power or torque being delivered to the rear wheel by gears, chains or a worm, the overcoming of the rolling resistance, the lifting up-hill of the unit and the drawbar resistance, as well as in regard to how those factors fluctuate.

Let us first consider the tractor statically. In Fig. 13, we have the weight of the tractor concentrated at its center of gravity, a height $H$ from the ground. This total weight is distributed front and rear in values of $w$ and $W$. When on the level, the perpendicular through the center of gravity divides the wheelbase line into the two parts $x$ and $k—x$, from which we develop:

$$Wx = w\ (k - x) \text{ or } Wx + wx = wk; \text{ and } x = wk/(W + w) \quad (6)$$

$$(k - x) = k - wk/(W + w) \text{ and } (k - x) = Wk/(W + w) \quad (7)$$

This distribution of weight is the same for forward motion of the tractor when there is no rolling resistance and no load. As soon as resistance is offered underneath the wheels, as is always the case, the weights change in value from $W$ and $w$ as follows:

The transmission chains, gears or worm must exert added power to advance the tractor and, therefore, we have a lifting tendency on the front end of the tractor. If $C$ is the chain pull and $G$ the sprocket diameter, the tangential reaction at the ground below the rear wheel would be equal to the torque $C \times G/2 = E \times R/2$ and the lift action $x$ at the front wheel would be $xk = CG/2$. Hence

$$x = CG/2k = \text{Torque} \div 2k = w_3 \quad (8)$$

This value can be understood better when we think of great resistance to translation; that is, to a point where forward motion is impossible. After a period of wheel slippage that buries the tractor, the chain then tends to wind on the sprocket and unwind on the gear, revolving the tractor about the rear axle as a center with a tendency toward overturn. The amount of this lift is as stated in formula (8), if it is within the capacity of the engine. The inertia of moving parts will add to or subtract from the value in accordance with the fluctuations in the rate of travel.

Let us suppose the case of a tractor running over medium soil and capable of a drawbar pull just sufficient to haul four similar outfits behind it. Then we have a rough measure of this resistance, in lieu of some better means of dynamometer measurement. We can specify the rolling resistance, $R\ R$, in terms of $D$ in either case, or we would have a coefficient in percentage of $D$. Take the four outfits mentioned above; the coefficient would be

$$D/4 = 0.25\,D. \quad D/4 \times R/2 = CG/2 = xk.$$
$$x = DR/8k \text{ or } 0.25\,DR/2k = 0.125\,DR/k = w_3 \quad (9)$$

The relation of tractor-wheel diameter with respect to rolling resistance was developed in some degree by A. F. Moyer,[5] but much work still remains to be done on this subject.

If, in addition, we have a developed load on the drawbar, the actual lift exerted on the front of the tractor

[5] See S. A. E. Transactions, vol. 13, part 1, p. 405.

Fig. 12—Three 45-Hp. Tractors Pulling 55 Plow Bottoms

will again be resisted by the front weight at the end of its lever arm. The point about which the rotative action takes place can be considered as at the ground contact of the drive-wheel rim. If the resistance exerted by the drawbar amounts to a true anchor, one of two things occur. The engine will be killed by the reaction, in the endeavor to lift off the front weight $w$, or there will be a combination effect of a slight backing of the tractor and a lifting off the ground of the front. The danger of overturn depends on the sum of the reactions of steady power and inertia of the moving parts and the slope of the ground. In any event, the rolling resistance and drawbar reaction during translation tend to lighten the ground load under the front wheel and transfer these loads to the rear. In Fig. 13 we see the development of the formula by which we can calculate the drawbar-lift value. In the plow and the drawbar values we have

$$(h + d/2) : L :: x_1 : (L + a) \text{ or}$$
$$x_1 = [(h + d/2)\ (L + a)] \div L$$
$$x_2 = (x_1 - d/2) \cos \beta.$$

Combining we have

$$x_2 = [\{ (h + d/2)\ (L + a) \div L \} - d/2] \cos \beta \qquad (10)$$

Then the lift in front $= D\, x_2 = w_2$ or

$$w_2 = D/k\, [\{ (h + d/2)\ (L + a) \div L \} - d/2] \cos \beta \qquad (11)$$

Again, because we are working on compressible material, the actual supporting surface in translation is

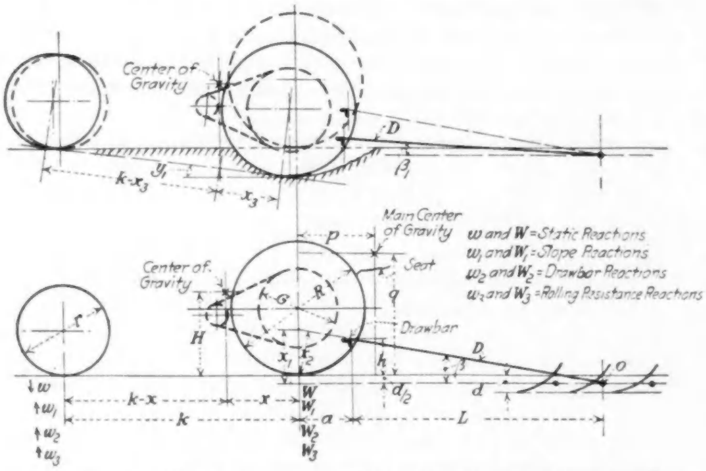



FIG. 13—DIAGRAM SHOWING THE VARIOUS REACTIONS IN THE TRACTOR

ahead of the vertical center-line through the rear axle by an amount which tends to set forward both support points, but not the center of gravity of the tractor, thus increasing still further the net weight carried by the supporting earth beneath the rear wheels.

### REACTIONS ON A SLOPE AND UP-HILL

To study the effect of slope in the direction of travel on the distribution of the tractor weight under the points of wheel contact, let us refer to Fig. 14, using an engine angle $\gamma$ with the horizontal; the wheelbase is as before, $a\ e$. On the level, the distribution of weight fore-and-aft due to the total weight at the center of gravity is inversely proportional to the distance $a\ c$ and $a\ e$.

To ascertain the relative value of these weights, it is necessary to develop a formula in accordance with the way the wheelbase line, $a\ e$, is divided by the projection of a perpendicular through the center of gravity, as at the point $d$, in Fig. 14. We then have $w_1$ the new weight at $a$, and $W_1$, at $e$ such that

$$W_1 + w_1 = W + w \text{ or, } W_1 = w + w - w_1 \text{ and}$$
$$w_1 = W + w - W_1 \qquad (12)$$

also,

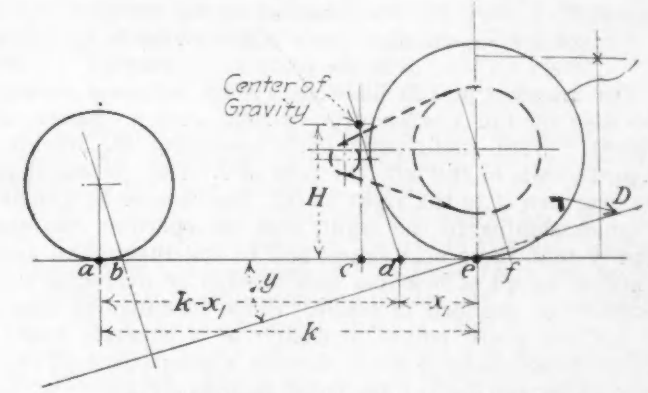



FIG. 14—EFFECT OF SLOPE ON THE TRACTOR

$$W_1 ad = W_1 de;\ ad = ac + cd \text{ and } de = ce - cd$$
$$w_1\ (ac - cd) = W_1\ (ce - cd)$$

Substituting the value of $a\ c$ and $c\ e$ in terms of $W$, $w$ and $k$

$$w_1\ (Wk/W + w) + w_1 H \tan \alpha = W_1\ (wk/W + w - H \tan y) \qquad (13)$$

Substituting values from formulas (12) and (13), first for $w_1$ and then for $W_1$ and reducing, we get

$$W_1 = W + (W + w/k)\ H \tan y \qquad (14)$$
$$W_1 = w - (W + w/k)\ H \tan y \qquad (15)$$

After interpreting these formulas, we find that the change in weight front and rear is directly dependent on the size of the angle of slope and inversely proportional to the wheelbase $k$. The original front weight decreases and the rear weight increases when going up-hill by the increments shown. Also, the higher the center of gravity is, the greater the change will be.

### CROSS-THE-FURROW SLOPES

We still have to review the reactions covering the cross-furrow hillside-actions, the turning moment met by the front wheels in steering due to pulling off-center, the effect of the operator in the seat, and the like. With what has gone before, these forces and reactions can be determined readily. We can note the tendency to slip down-hill in Fig. 15 and how the weights front and rear which, on the level ground can be considered as divided equally between the two front and the two rear wheels, on hillsides are split into further differences in the same general manner as the longitudinal slope reactions.

Let $W$ denote the weight of the tractor drive-wheel end and $W/2$ be the normal weight on each rear wheel. Let $N$ equal the tractor width over the drive-wheel rims and $H_1$ the gravity-center height between these wheels. Then the load on the down-hill wheel will be to the load on the up-hill wheel as the perpendicular divides the ground line and we have:

$$W_u \times N/2 + H_1 \tan \delta = W_d \times N/2 - H_1 \tan \delta \qquad (16)$$

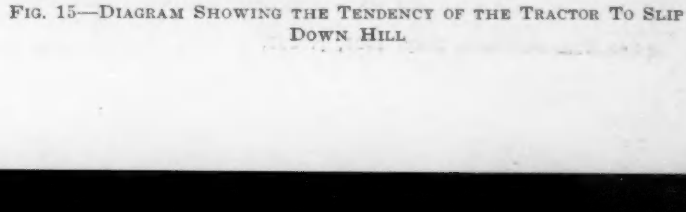



FIG. 15—DIAGRAM SHOWING THE TENDENCY OF THE TRACTOR TO SLIP DOWN HILL

$$W_u + W_d = W \quad \text{(For the rear of the tractor)} \tag{17}$$

$$w_u \times n/2 + H_s \tan \delta = w_d \times n/2 - H_z \tan \delta \tag{18}$$

$$w_u + w_d = w \quad \text{(For the front of the tractor)} \tag{19}$$

The drawbar pull $D$ likewise has its influence on slip sidewise of the tractor drive-wheels, and a change in weight. With small plow units operating on hillsides sloping down to the left, the pull in $D$ tends to stabilize the overturn if to the right of $O$. The reverse is true on hillsides sloping to the right, and the operator can see readily that these side slopes add to the theoretical and practical range of hitching in a manner to overcome the tendency of the load to range. Since formulas at times do not bring out points so clearly as a concrete example, an endeavor is made to develop a comparison of two tractors by the use of formulas to indicate the relative value of these tractors as to stability. We have selected two popular tractors which were reported upon in the University of Nebraska tests.[6] Their weight data as given are supplemented by available data as to their centers of gravity and the like, as shown in Table 7.

TABLE 7—COMPARISON OF TWO TRACTORS

| Data | Tractor *A* | Tractor *B* |
|---|---|---|
| Total Weight, lb. | 2,710 | 5,708 |
| Static Front Weight, lb. | 1,075 | 1,890 |
| Static Rear Weight, lb. | 1,635 | 3,818 |
| $H$, in. | 28.0 | 33.5 |
| $p$, in. | 5 | 30 |
| $q$, in. (Height of Center of Gravity of the Man on the Seat) | 41 | 50 |
| Engine Speed, r.p.m. | 1,006 | 550 |
| Drawbar Pull at the Speed Given, lb. | 1,428 | 1,850 |
| Drawbar Horsepower | 8.26 | 15.65 |
| Tractor Speed, m.p.h. | 2.17 | 3.17 |
| Wheelbase, in. ($= k$) | 63 | 92 |
| Rear Axle to Drawbar, in. ($= a$) | 12.0 | 20.5 |
| $h$, in. | 15.0 | 15.0 |
| $L'$, in. | 89.5 | 99.0 |
| $d$, in. | 6.0 | 6.0 |
| $r$, in. | 27.5 | 36.0 |
| $R$, in. | 42.0 | 54.0 |
| $G$, in. | ... | 31.0 |
| $k - x$, in. | 38.0 | 61.5 |
| $x$, in. | 25.0 | 30.5 |

[7] Two-bottom for Tractor *A* and three-bottom for Tractor *B*.

To put the two on a more comparable basis we will interpolate the drawbar pull with respect to the speed in miles per hour. The ratio will be the same whether we increase the speed of one or decrease the speed of the other, or whether we assume an intermediate desired speed and bring both to that speed. Then (1850 lb. $\times$ 3.17) $\div$ 2.17 = 2700 lb. This is a figure closely in line with the maximum pulled per the test; hence, this figure is practical to use. Applying formulas (6) and (7), we find the distances $x$ and $k—x$ for each.

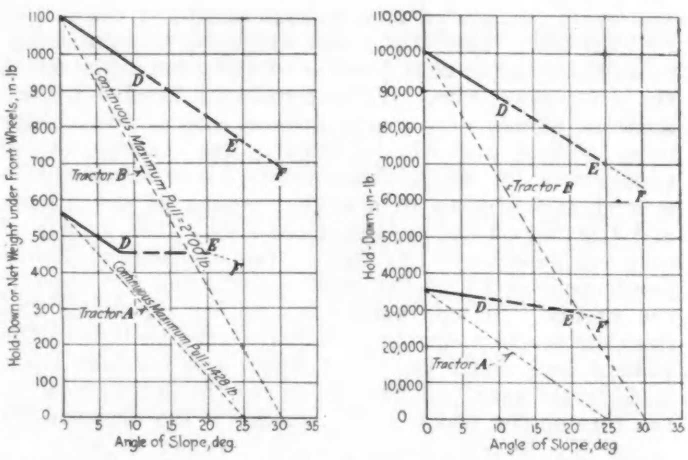



FIG. 17—COMPARISON OF TWO TRACTORS UNDER A CONSTANT OPERATING LOAD AT RATED ENGINE SPEED

We can use formulas (8) or (9) for rolling resistance and we shall assume that this is such as to be equal to one-quarter of the drawbar pull, or 376 and 675 lb. respectively; hence, formula (9) applies and we have 119 lb. and 198 lb. as the respective lifts on the front wheels of the two tractors due to the rolling resistance being overcome. Formulas (10) and (11) are used for the drawbar lift, and formulas (14) and (15) for the changes due to slope. The man-in-the-seat data are obtained from the formula.

TABLE 8—COMPARISON REGARDING SAFETY AGAINST OVERTURNING

| | Tractor *A* | Tractor *B* |
|---|---|---|
| Weight, lb. | | |
| Tractor | 2,710 | 5,708 |
| Operator | 150 | 150 |
| Drawbar Pull (Vertical Reaction= $D \sin B$) | 280 | 484 |
| Total | 3,140 | 6,342 |
| $D$, lb. | 1,428 | 2,700 |
| $B$ | 11 deg. 19 min. | 10 deg. 18 min. |
| $\sin B$ | 0.1962 | 0.1788 |

| Lifting Effect, lb., due to | Tractor *A* Basis, Deg. Level | 6 | 12 | 18 | 24 | Tractor *B* Basis, Deg. Level | 6 | 12 | 18 | 24 | 30 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Slope | 0 | 126 | 250 | 372 | 490 | 0 | 217 | 432 | 642 | 846 | 1,040 |
| $R$ $R$ (Rolling Resistance) | 119 | 119 | 119 | 119 | 119 | 198 | 198 | 198 | 198 | 198 | 198 |
| Drawbar | 384 | 384 | 384 | 384 | 384 | 540 | 540 | 540 | 540 | 540 | 540 |
| Operator | 12 | 22 | 33 | 44 | 56 | 49 | 58 | 67 | 78 | 90 | 104 |
| Total | 515 | 651 | 786 | 919 | 1,049 | 787 | 1,013 | 1,237 | 1,458 | 1,674 | 1,882 |
| Net Weight, lb. | | | | | | | | | | | |
| Front | 560 | 424 | 289 | 156 | 26 | 1,103 | 877 | 653 | 432 | 216 | 8 |
| Rear | 2,580 | 2,716 | 2,851 | 2,984 | 3,116 | 5,239 | 5,465 | 5,689 | 5,910 | 6,126 | 6,334 |

$$w_4 = 150 \text{ lb. } (p + q \sin y/k) \tag{20}$$

Then the net weight on the ground under the front wheel equals the original weight less the sum of the $RR$ or rolling-resistance lift, drawbar-pull lift, slope lift and man lift. These weights are shown with respect to slopes in Table 8 and also the relative inch-pounds of advantage at the various slopes. However, to get the real relations we should solve the data with regard to the safety against overturn per 1000-lb. drawbar pull.

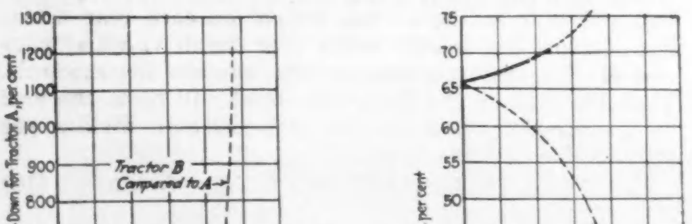



FIG. 16—RELATIVE STABILITY OF THE TRACTOR AGAINST OVERTURNING

[6] See THE JOURNAL, May, 1921, p. 391.

TABLE 9—DRAWBAR HORSEPOWER LOSSES DUE TO WEIGHT-LIFT UP-SLOPE

| | Basis, deg. | | | | |
|---|---|---|---|---|---|
| Loss Due to Weight-Lift, hp. | Level | 6 | 12 | 18 | 24 |
| Tractor *A* | 0 | 2.28 | 4.54 | 6.74 | 8.78 |
| Tractor *B* | 0 | 4.18 | 8.32 | 12.37 | 16.13 |
| Drawbar Pull, hp. | | | | | |
| Tractor *A* | 8.27 | 5.99 | 3.73 | 1.53 | —0.51 |
| Tractor *B* | 15.63 | 11.45 | 7.31 | 3.26 | —0.50 |

Fig. 16 gives these values in definite figures on a comparable basis. As a matter of fact, neither of these two outfits could negotiate the angles calculated, as we have assumed the drawbar pull near to its maximum when operated on the level; but, by assuming the power developed to be sufficient to carry it through, we can see the relations.

To get the true relations, then, we must limit the discussion to those angles up to which the tractor can pull its rated load, or to two and three plows respectively. Assuming that this machine is working in average light soil of 450 lb. per 14-in. bottom, at 2.17 m.p.h., we have drawbar pulls of 900 and 1350 lb. respectively as minima usable loads for plowing. Checking off the horsepower required to negotiate the different angles of slope, with the weights of tractor, plow and man, at the constant rate of speed chosen, we obtain the values shown in Table 9.

From this we can interpolate and find that these units can negotiate a slope of about 8 deg. for Tractor *A* and 11.25 deg. for Tractor *B*, when pulling their rated plow loads. However, with continually reduced loads these units will be able to climb increased slopes until the tractors have no available drawbar pull. This maximum angle is 20 deg. for Tractor *A* and 24 deg. for Tractor *B*. The net load under the front wheel, however, persists in being positive until 24 deg. for Tractor *A* and until 30 deg. for Tractor *B*. Fig. 17 illustrates the actual ratios in percentages and per 1000-lb. drawbar pull.

Formulas (16) and (17) apply not only to hill slopes but likewise to tractors operated in the furrow. When thus operated, the average tractor tilts out of plumb 6.5 deg. for two-plow units; 6 deg. for three-plow and 5.5 deg. for four-plow units. When in action, this tilting throws a heavier load on the furrow wheels than normally is imagined. Applying formulas (16) and (17) to the two outfits operating on level ground, but having one wheel in the furrow, we have the values shown in Table 10.

While the above analysis might be refined more closely and correctly, the main features developed will clear up and segregate the main influences affecting the several actions and reactions the net sum of which at any instant can be ascertained approximately by the simple formulas developed. We believe that the solutions to the problems propounded can now be determined readily for every case involving static relations, rolling resistance, distribution of weights, balance, choice of hitch and position of operation, whether on land or in the furrow; also, that the analysis presented gives us a definite method of attack for the more correct solution of the proper hitching-point, as well as being a study relating to lug design.

TABLE 10—COMPARISON ON LEVEL GROUND WITH ONE WHEEL IN THE FURROW

| | Tractor *A* | Tractor *B* |
|---|---|---|
| Width over Tires, in. | 62 | 56 |
| Values of *CG*, in. | 28.0 | 33.5 |
| Rear Weight, lb. | 2,580 | 5,239 |
| Weight on Furrow Wheel, lb. | 1,390 | 2,949 |
| Weight on Land Wheel, lb. | 1,190 | 2,290 |
| Difference, lb. | 200 | 659 |

## THE DISCUSSION

E. R. WIGGINS:—It was mentioned that various factors remain the same when going up a hill. Would not the weight on the rear wheels be more?

O. B. ZIMMERMAN:—The weights change back according as the center line between the points of wheel contact is divided by the perpendicular dropped from the center of gravity as described in the paper. It is remarkable what a wide range of change there is. For instance, take a small tractor weighing 2710 lb., place a man in the seat who weighs say 150 lb., add to this the other component, which is the down pull of the drawbar, and the total combined weight is then 3140 lb. When on the level and pulling the maximum, according to the Nebraska tests, 2580 lb. of the 3140 lb. is on the rear wheel and only 560 lb. on the front wheel. When the tractor is standing idle, we have 1070 lb. to begin with, which now has dropped to 560 lb. when the load is being pulled. Supposing there were engine power enough to carry the tractor up the 24-deg. slope, the net weight on the front wheel at 24 deg. is only 26 lb. and the weight on the rear wheels 3116 lb. One can see readily that when such an angle is reached it takes very little to turn the tractor over. Consider the heavier tractor, which weighed 5708 lb. according to the Nebraska tests. The man weighed 150 lb. Adding the down pull of the drawbar gives a total of 6342 lb. When we start on the level we have 1103 lb. on the front and 5239 lb. on the rear wheels; that is, in action. When we get up to the 30-deg. slope, we have an 8-lb. hold down on the front and 6334 lb. on the rear wheels. When we divide that rear-wheel weight according to the right and the left wheels and according to the slope that there may be on the hillside, we are aware that we have some tremendous loads under each wheel. The interesting point there in connection with lugs is that the drawbar pull backward is small as compared with the push downward underneath the wheels; so the lugs do not need to be so long as many manufacturers are making them.

G. D. JONES:—In regard to the plow test, what form of hitch did Mr. Zimmerman use in making the tests?

MR. ZIMMERMAN:—They were all made with the same hitch; but, at that time, we did not realize that the differences in hitching were so great between one position and another.

MR. JONES:—Did you have a hitch on the order of the old A-type?

MR. ZIMMERMAN:—We used the A-type hitch or a regular solid drawbar.

MR. JONES:—Were no tests made of various forms of hitch?

MR. ZIMMERMAN:—No. That is one of the problems that either agricultural engineers or the Society should study.

MR. JONES:—We experimented some 2 years on hitches, and found decided changes with different types of hitch. I am wondering what reaction one gets at various changes.

MR. ZIMMERMAN:—One would need to work out a theoretical hitch and work on that as a basis. There is much to be developed on this subject. This analysis has been based on practical experience and on definite deductions which for the particular moment were correct, but we still do not know the cross-furrow reaction, styled *R*, as represented for various peaks.

PROF. J. B. DAVIDSON:—I take it that the horse-drawn plow is better designed, because the cross-furrow reaction, styled *R*, is carried on the plow carriage and results in a lighter draft; in other words, the tractor plow is not scientifically designed.

MR. ZIMMERMAN:—That one wheel in the corner of the furrow takes the thrust and we thus change the thrust from a sliding friction, which we have to a large extent in the tractor plow, to a rolling friction. When we get into large units, such as 3, 4, 5, 8 or 12 bottoms, the furrow wall would not take the thrust component; so, we must set the tractor over to take care of the draft. But if we do pull on the slant in that way, according to the plow friction, we take it up in thrust on the engine bearings unless we provide for rolling friction there by using roller bearings or something of that kind.

PROFESSOR DAVIDSON:—I am glad Mr. Zimmerman mentioned that because it does not seem right to carry the side-thrust on a land-side with an increase in the resistance due to friction. We can demonstrate very easily with a horse-drawn plow and a dynamometer that this may mean about a 25-per cent increase in the draft.

I think it is a matter of record that some of the accidents due to tractors overturning have happened when the tractors were detached from their loads. This is explained by the fact that the tractor revolves about the rear axle when the drive-wheels become fixed. If the load is attached and the hitch is below the axle, the tractor could not very well turn over because, so long as the center of moments coincides with the center of the axle, the drawbar pull tends to keep the front end of the tractor down.

# CAMBER AND GATHER RELATIONSHIPS

(*Concluded from page* 92)

TABLE 4—GATHER FOR DIFFERENT WHEEL DIAMETERS AND AREAS OF CONTACT

| *D*, Wheel Diameter, In. | *L*, In. | | | | |
|---|---|---|---|---|---|
| | 6 | 7 | 8 | 9 | 10 |
| 30 | 0.180 | 0.210 | 0.240 | 0.270 | 0.300 |
| 32 | 0.192 | 0.214 | 0.256 | 0.288 | 0.320 |
| 34 | 0.204 | 0.238 | 0.272 | 0.306 | 0.340 |
| 36 | 0.216 | 0.252 | 0.288 | 0.324 | 0.360 |
| 38 | 0.228 | 0.266 | 0.304 | 0.342 | 0.380 |
| 40 | 0.240 | 0.280 | 0.320 | 0.360 | 0.400 |
| 42 | 0.252 | 0.294 | 0.336 | 0.378 | 0.420 |

from which we can prepare Table 4, which gives values of gather in inches.

### CORRECT GATHER FOR FRONT WHEELS

Table 4 can be summarized by the following rule: The correct gather for front wheels, *G*, is obtained by multiplying the diameter of the tire in inches by the length of the area of contact in inches and dividing by 1000; that is, $G = DL/1000$.

The procedure in making use of Table 4 is to see first that the air pressure and load are normal, and that the front wheels rest upon a smooth flat surface; then to measure the length of the area of contact. The length of the area of contact can be determined accurately by first jacking up the wheel and then lowering it carefully upon a piece of white paper covered with a piece of carbon paper or multigraph ribbon. When the wheel is lifted again, a print of the area of contact will be found on the white paper, from which the length *L*, in inches, can be obtained. If, for lack of equipment, the above instructions cannot be followed, an approximation of the lengths of the areas of contact can be taken from Table 5, which gives values of *L* for various sizes of pneumatic tires under S.A.E. Standard loads and inflation-pressures. Second, jack up both front wheels and turn them, scribing a fine line near the center of the tread, as indicated in Fig. 4. Third, refer to Table 4. For example, suppose that the tire is 32 in. in diameter and the length of the area of contact is 8 in. A glance at Table 4 shows that the gather should be 0.256 in. or slightly more than ¼ in.; that is, the distance between the tread lines should be about ¼ in. greater behind than in front. Scribing the lines on the treads eliminates any error due to wobbling of the wheel, but the bearings and the bushings should be adjusted closely.

It should be understood clearly that the above method applies only to the average case in which *S* equals approximately 500 in. This is sufficiently accurate in nearly all cases. There are, however, a few cases in which the camber differs noticeably from the average and, therefore, *S* is noticeably greater or less than 500 in. For these special cases the formulas, rather than the tables, should be used.

The entire operation of checking the front-wheel alignment of a vehicle requires very little time and practically no expense. It is worth any owner's time to construct a suitable "tram" which can be made easily of a few pieces of wood, as indicated in Fig. 4. Public garages should be equipped with a tram for this purpose. The life of a front tire may be reduced many thousands of miles through a misalignment of ⅜ in.

TABLE 5—APPROXIMATE LENGTHS, *L*, OF AREAS OF CONTACT

| Fabric Tire Size, In. | Cord Tire Size, In. | S.A.E. Standard Air Pressure, Lb. Per Sq. In. | S.A.E. Standard Load, Lb. | *L*, Length of Area of Contact, In. |
|---|---|---|---|---|
| 30x3 | .... | 45 | 375 | 6.3 |
| 30x3½ | .... | 55 | 570 | 6.6 |
| .... | 30x3½ | 50 | 600 | 6.5 |
| .... | 32x3½ | 50 | 600 | 6.4 |
| .... | 31x4 | 60 | 850 | 6.9 |
| 32x4 | .... | 65 | 815 | 7.0 |
| .... | 32x4 | 60 | 850 | 7.0 |
| .... | 33x4 | 60 | 850 | 6.8 |
| .... | 32x4½ | 70 | 1,200 | 7.6 |
| .... | 34x4½ | 70 | 1,200 | 7.6 |
| .... | 35x5 | 80 | 1,700 | 8.6 |
| .... | 36x6 | 90 | 2,200 | 9.0 |

# Current Standardization Work

ALTHOUGH the work of the various Divisions during the last 2 months was concentrated on those recommendations that were presented for approval at the Standards Committee Meeting on June 20, many subjects have received consideration which indicates that the reports to be presented at the Standards Committee Meeting in January 1923 will require much time and effort during the last half of 1922.

Eleven Division and Subdivision meetings were held during May and June as shown in the accompanying table. No Division meetings have been scheduled for the summer months, pending a review of the work now in progress and the formulation of sufficient Subdivision reports to warrant calling the Division members together.

The discussion which was presented at the Standards Committee meeting on June 20 and an account of the official action taken on the various reports will be printed in full in the August issue of THE JOURNAL. A brief summary of the action taken, however, is printed in this issue in the account of the Summer Meeting beginning on p. 1.

| | |
|---|---|
| Axle and Wheels Division | May 2 |
| Electric Vehicle Division | May 10 |
| Lighting Division | May 3, June 2 |
| Motorboat Lighting Subdivision | May 2 |
| Non-Ferrous Metals Division | May 1 |
| Parts and Fittings Division | May 9 |
| Passenger Car Division | May 2 |
| Passenger-Car Hubs Subdivision | April 17 |
| Screw-Threads Division | May 1 |
| Springs Division | May 5 |
| Storage-Battery Division | May 5 |

### BALL BEARINGS

A series of meetings was held on April 27 and 28 at which the subject of international standardization of ball bearings was carefully considered and tentative proposals adopted which have been submitted to England, Sweden, Germany and other European countries.

The Committee on Information, appointed by the American Sectional Committee on Ball Bearings, met on the morning of April 27 to consider the report of O. R. Wikander, who had just returned from a trip abroad, covering results of his discussions with European ball-bearing representatives as to international standardization. The Subdivision reported to the Sectional Committee on the afternoon of April 27, its report being approved as submitted and informally referred to the ball-bearing committees of the American Society of Mechanical Engineers and the Society of Automotive Engineers, the sponsor bodies for the Sectional Committee, for consideration on the morning of April 28. The Division of the Society's Standards Committee and the committee of the American Society of Mechanical Engineers approved the corner radii proposed by the German Ball Bearing Committee and favored the other proposals as submitted by the Sectional Committee which are given hereinafter. The proposals were referred back to the Sectional Committee in the afternoon of April 28 and definite proposals were prepared which were subsequently submitted, as stated, to other countries interested in establishing international ball-bearing standards. The proposals of the American Sectional Committee on Ball Bearings are as follows:

(1) That the outside diameters proposed by the German Ball Bearing Committee be accepted for the Light, Medium and Heavy Series

(2) That the widths of the bearings covered by the above Series be retained for those sizes for which International uniformity exists at the present time, that is, up to and including 110-mm. bores for the Light Series, 95-mm. bores for the Medium Series and 85-mm. bores for the Heavy Series; but that above these sizes, the width of each bearing shall be approximately 80 per cent of the radial difference between the bore and the outside diameter, which conforms closely to Swedish practice

(3) That if an Extra-Light Series, which is not favored, should be adopted, the widths of such bearings should be made approximately 80 per cent of the radial difference between the outside diameters and the bores

(4) That the corner radii proposed by the German Ball Bearing Committee be adopted

(5) That the width tolerances adopted by the Society of Automotive Engineers be adopted instead of the very close German width tolerances

The German tolerances for bores and outside diameters correspond so closely to the tolerances adopted by the Society of Automotive Engineers that they are identical for all practical purposes.

No further action will be taken by the American Sectional Committee on Ball Bearings until information is received from abroad as to whether these proposals are satisfactory and, if not, what revised proposals would meet with approval.

### BASES, SOCKETS AND CONNECTORS

At the meeting of the Lighting Division on May 3 a Subdivision was appointed to investigate thoroughly the present S. A. E. Standard for Bases, Sockets and Connectors, p. B4 of the S. A. E. HANDBOOK, and if advisable to recommend revisions to meet the objections that have been submitted against this standard by motor-truck engineers. The Subdivision will also consider the standardization of this type of equipment for motorboats. The personnel of the Subdivision appointed is:

| | |
|---|---|
| C. E. Godley, *Chairman* | Edmunds & Jones Corporation |
| A. K. Brumbaugh | Autocar Co. |
| J. T. Caldwell | National Lamp Works |
| B. H. Kenyon | Providence Base Works |
| J. C. Stearns | Culver-Stearns Mfg. Co. |
| G. A. Walters | Chicago Electric Mfg. Co. |
| Ernest Wooler | Cleveland Automobile Co. |

### BESSEMER STEELS

As a result of a request submitted by the International Harvester Co., the addition of bessemer steels to the S. A. E. Steel Specifications was considered at the last meeting of the Iron and Steel Division. It was pointed out that many users feel that including bessemer steel specifications in the S. A. E. Standard would not be advancing the art, but it was thought that separate specifications might be desirable to meet requirements for agricultural implements. It was considered advisable, however, to obtain definite evidence supporting the need of bessemer steel specifications before taking final action.

After general discussion, it was decided to reserve the 1400 series for bessemer steels with the understanding that this would not commit the Society to formal approval of such compositions or their uses, and to recommend that the implement manufacturers cooperate with each other toward deciding upon a limited series of bessemer steels, in which work the Iron and Steel Division will assist as an advisory body.

### COTTER-PINS

At the meeting of the Parts and Fittings Division on May 9 it was stated that trouble had been experienced in obtaining S. A. E. Standard cotter-pins due to the drill numbers for the holes as given in the present S. A. E. Standard, p. C7 of the S. A. E. HANDBOOK, being too large for the manufacturers' standard sizes of cotter-pin. Cotter-pin

wire sizes are determined from the drill size for the holes in which they are to be used rather than from the actual diameter of the wire. Thus, according to the present S. A. E. Standard, a purchaser would have to buy oversize cotter-pins to fit S. A. E. Standard holes. It was suggested that the present standard should be revised by eliminating the 7/64 and 9/64-in. cotter-pin sizes and by changing the 11/64-in. size to 5/32 in. and the 13/64-in. size to 3/16 in. and by inserting the actual diameter of the wires and decreasing the drill number for the holes. This led to discussion as to how tight a cotter-pin fit is desired, but no definite recommendation was agreed upon.

The subject is to receive further consideration, particularly in regard to what effect changes such as suggested would have on other S. A. E. Standards.

### Bolt Heads and Rivets

As the Sectional Committee on Bolt, Nut and Rivet Proportions, sponsored by the American Society of Mechanical Engineers and the Society, is working on the standardization of bolt heads and rivets because they are of general interest to other than the automotive industry, it has been decided to hold the standardization of these subjects by the Screw-Threads Division in abeyance.

### Engine Mufflers

The reasons in support of the negative votes cast by Society members against the adoption of the S. A. E. Standard for Mufflers, p. A13 of the S. A. E. Handbook, were reviewed at the last meeting of the Engine Division. It was thought that no revisions should be made in the present standard until it shall have been applied in actual practice and further information obtained as to its practicability.

### Engine Support Arms

At the meeting of the Engine Division on April 17 the further standardization of engine support arms was discussed with special reference to sub-frame construction. It was considered inadvisable to standardize on a sub-frame construction unless there shall be a clear demand for such action. The matter is to be referred to engine and motor-truck manufacturers for comment.

### Felt Specifications

Criticisms of the proposed felt specifications, resulting from circularizing the tentative report of the Subdivision on Felt, published on p. 435 of the May issue of The Journal, were discussed at the Parts and Fittings Division meeting held on May 9. It was the thought that these criticisms should be sent to the members of the Subdivision for further study and that laboratory tests should be specified in the recommendation. Reference was made to the more or less satisfactory results of tests that have been developed and enforced by a number of automobile companies and it was felt that the Subdivision should obtain as much information as possible regarding these tests.

### Head-Lamp Brackets

At a meeting of the Lighting Division on May 3 the suggestion that the present S. A. E. Standard for Fork-Type Head-Lamp Brackets should be cancelled was discussed. It was brought out that this type of head-lamp mounting is still used to a considerable extent on motor trucks and motorcycles and that the present standard should be retained as a guide to passenger-car builders using it.

### Fuel and Lubrication Pipe Fittings

At the meeting of the Parts and Fittings Division on May 9 it was stated that the present S. A. E. Recommended Practice for Flared-Tube Type of Fuel and Lubrication Pipe Fittings, p. C46 of the S. A. E. Handbook, was being largely replaced by the compression type of coupling, particularly on truck and tractor installations, owing to the severe vibration. It was decided, however, to retain this standard temporarily at least.

As it was thought that the Division should standardize a series of compression-type couplings, a Subdivision was appointed to prepare a report that will include a list of sizes of compression-type coupling which will be submitted to the industries for comment.

The present S. A. E. Recommended Practice for Soldered-Type Couplings, p. C47 of the S. A. E. Handbook, was reviewed. As this specification is used principally by motorcycle manufacturers and has evidently proved satisfactory, it was decided to retain it without revision.

### Poppet Valves

A Subdivision has been appointed to extend the present S. A. E. Standard for Poppet Valves, p. A3 of the S. A. E. Handbook, so as to include the dimensions for the valve head. The present standard specifies only the port diameter and the corresponding valve and stem diameters.

### Passenger-Car Front-Axle Hubs

At the Division meeting held on May 2 five sizes of passenger-car front-axle hub were studied in connection with ball and roller-bearing layouts that had been submitted. The ball-bearing layout included a new series of inch dimension ball bearings that have the same bores as the proposed roller-bearing series, as it is thought that the ball-bearing manufacturers will be willing to develop such a series. It was decided that the recommendation should include the spindle lock-nuts and washers and hub-cap threads.

To facilitate the work of the Subdivisions, a ball-bearing as well as a roller-bearing subcommittee was appointed with axle, wheel and passenger-car representation on each. The subcommittees are to prepare jointly a single proposal for ball and roller-bearing applications, this to be reported to the Subdivision so that flange diameters, spoke widths and flange-bolt dimensions may be added. The proposal will then be circularized among the passenger-car and parts manufacturers for comment.

BALL-BEARING SUBCOMMITTEE PERSONNEL

| | |
|---|---|
| F. W. Gurney, *Chairman* | Gurney Ball Bearing Co. |
| R. S. Begg | Jordan Motor Car Co. |
| H. E. Brunner | S. K. F. Industries, Inc. |
| E. R. Carter, Jr. | Fafnir Bearing Co. |
| L. A. Cummings | Standard Steel & Bearings, Inc. |
| C. S. Dahlquist | Eaton Axle Co. |
| A. M. Dean | Templar Motors Co. |
| F. G. Hughes | New Departure Mfg. Co. |
| A. M. Laycock | Sheldon Axle & Spring Co. |
| A. L. Putnam | Detroit Pressed Steel Co. |
| O. J. Rohde | Wire Wheel Corporation of America |
| L. M. Stellman | H. H. Franklin Mfg. Co. |
| H. Vanderbeek | Timken Detroit Axle Co. |
| Andrew S. VanHalteren | Motor Wheel Corporation |

ROLLER-BEARING SUBCOMMITTEE PERSONNEL

| | |
|---|---|
| T. V. Buckwalter, *Chairman* | Timken Roller Bearing Co. |
| R. S. Begg | Jordan Motor Car Co. |
| L. W. Close | Bock Bearing Co. |
| C. S. Dahlquist | Eaton Axle Co. |
| A. M. Dean | Templar Motors Co. |
| G. W. Dunham | Savage Arms Corporation |
| C. T. Hagenlocker | Wright Roller Bearing Co. |
| A. M. Laycock | Sheldon Axle & Spring Co. |
| A. L. Putnam | Detroit Pressed Steel Co. |
| O. J. Rohde | Wire Wheel Corporation of America |
| R. G. Schaffner | Bower Roller Bearing Co. |
| L. M. Stellman | H. H. Franklin Mfg. Co. |
| H. Vanderbeek | Timken-Detroit Axle Co. |
| Andrew S. VanHalteren | Motor Wheel Corporation |

### Storage-Battery Monoblock Containers

Tentative specifications for storage-battery monoblock containers were approved at the meeting of the Storage-Battery Division on May 5. These specifications are intended as an extension of the present S. A. E. Standard for Storage

(*Concluded on page* 124)

# THE SUMMER MEETING

(*Concluded from page 8*)

J. G. Vincent supplemented Mr. Crane's remarks with a discussion of the relative merits of the three types of engine construction: L-head, I-head and overhead-camshaft. Experiments had been made by the Packard company on all three types. In the overhead-camshaft type the lack of accessibility had been the principal disadvantage. From the practical point of view he considered that the choice lay between the L-head and the pushrod-operated overhead-valve engine. Each of these types has its advantages, and both are equally quiet. He questioned whether it is worth while to put valves in the head to get better volumetric efficiency. He recommended that particular attention be paid to carburetion. In conclusion, he remarked that the overhead-camshaft engine cannot be produced in quantities because of its high cost and of the difficulty of securing quiet operation.

F. S. Duesenberg called attention to the design of an overhead-camshaft engine constructed so that the cylinder-head can be removed without interfering with the timing of the valves. R. E. Fielder spoke of the merits of the sleeve-valve engine. In concluding the discussion of the relative merits of the different types of engine construction, A. L. Nelson stated his preference for the valve-in-head engine, while Mr. Heldt summed up by predicting that the overhead-camshaft type will be confined to the higher-priced cars.

H. M. Crane then read his paper on a New System of Spring-Suspension for Automotive Vehicles. He indicated what the history of spring-suspension has been, but discussed only the conventional type of four-wheeled design, in which the front wheels are used for steering and the rear wheels for driving and braking. The problem of front-axle suspension is mentioned in the paper but that of proper rear-axle spring-suspension, especially for passenger cars, is discussed in detail because it is a much more difficult one.

Mr. Crane mentioned the advantages of the Hotchkiss drive for shaft-driven cars and some of its disadvantages.

W. C. Keys inquired whether Mr. Crane had made an investigation of the characteristics of cars using full-elliptic springs. Mr. Cravens made an allusion to the types of spring-suspension used in the Lanchester, the Lafayette and a number of other cars. Mr. Crane was of the opinion that full-elliptic springs do not afford sufficient resistance to torque reaction. The design he had adopted permitted the amount of longitudinal rigidity to be regulated. He emphasized the importance of rigidity against thrust and brake action.

In reply to Mr. Keys' inquiry as to the type of universal-joint used, Mr. Crane said he used a rigid universal-joint at the front end, while the rear-end joint was of the type in which rollers travel in a slotted member. This type of universal-joint is characterized by unlimited angularity together with great freedom of longitudinal movement. Mr. Crane referred to the popularity of the torque-arm drive and said that the chief objection to it is its great rigidity in absorbing torque reactions. He expressed the opinion that the use of cantilever springs necessitates the torque-arm construction. He was in favor of this type of construction for smooth roads. He said that another objection to the cantilever type of spring is the difficulty of supporting bodies owing to the overhang at the rear of the car. He emphasized the difficulty of making the rear frame strong enough to carry the rear weight, particularly in closed-body designs. He laid particular stress on the importance of frame stiffness rather than frame strength.

F. A. Bonham read his paper, The Automotive Engineer and Our Service Problem. He touched upon the work of the National Automobile Chamber of Commerce Service Committee and attributed the present condition to a misunderstanding of the service problem by the automotive industry generally. Mechanics were inadequately trained and were not equipped with the proper tools. To alleviate the present unsatisfactory state of affairs, he suggested that parts be made interchangeable as far as possible, that flat rates be charged for repairs, and that the necessity for special tools for certain makes of car be eliminated by standardization. He mentioned as the chief requirements, accessibility and simplicity and uniformity in the design of units and wearable parts. He particularly urged that the necessity for special and elaborate tools be eliminated, because the average service-station has not the time or the money or the space for them.

Mr. Bonham stated that the Service Committee is now running a series of tests on the road to determine which parts wear out soonest, and how long it takes to replace various parts; which parts are most prone to give trouble when there is divergence in manufacturing specifications or when in the hands of careless mechanics. The service problem is stated to be not one of design or of assembly and construction but rather of incompetence and ignorance on the part of the average repair-shop mechanic.

In discussing the points raised, T. J. Lítle, Jr., stressed the necessity for frequent conferences between the engineering and the service departments of the manufacturing companies. C. M. Manly thought that the problem is largely one of management and that complaints should be carefully investigated and cleared up. He also emphasized the necessity of cooperation between the different departments.

Mr. Bachman alluded to the matter of deterioration. He stated that the service end needs competent men who can give service and sales men a rational idea of what problems are of the most pressing importance.

Prof. W. K. Hatt presented an abstract of his paper on Highways. He described the organization of the Highway Research program that is now in progress under the auspices of the National Research Council, and mentioned the various bodies that are participating in its work throughout the Country. Slides were shown in setting forth the organization of the project, and illustrations were given of the equipment used and the methods of attacking the problem.

T. V. Buckwalter opened the discussion by stating that the kernel of the highway problem lies in banking the curves and cambering the tangents. H. W. Alden brought out a very interesting point when he said that a bad road is as bad for the road as for the vehicle; in other words, that the stresses of a vehicle on a bad road are as destructive to the road as to the vehicle. Professor Hatt agreed with this view and stated that in considering the highway problem we must evaluate the road in terms of transportation. He remarked that all roads im-

prove with age, because they settle. A new road is almost invariably a bad road.

Professor Hatt made a plea for the cooperation of automotive engineers in dealing with the problems that best the highway engineers. Primarily, the building and maintenance of good roads is not the concern of the automotive engineer, but it is very much to his interest to see that the Country is provided with a network of good roads, scientifically constructed so as to inflict the minimum wear or stress on automobiles and trucks. The abuse of roads by any one class of vehicles inevitably causes hardship to another class. Nothing can be achieved without the cooperation of automobile engineers and the National Research Council is doing its part by placing in their hands all the necessary information on the tractive resistance of roads to various types of vehicle, and on the relative efficiency and wearing qualities of different types of road.

### THE SPORTS PROGRAM

With unusually comprehensive and well-maintained sports facilities at White Sulphur Springs, it was anticipated that the sports program would be one of the outstanding features of the meeting. There were tournaments in baseball, golf, tennis and trapshooting, in addition to the aquatic meet and the field day. Inter-Section rivalry was always in evidence and added greatly to the interest in the program. A very fine collection of prizes rewarded the victors in the many contests, as well as a large percentage of the also-rans. These handsome prizes were made available through the generosity of over 200 firms in the industry who contributed to the prize fund.

#### GOLF TOURNAMENTS

Golf continues to be the most popular sport among the automotive engineering fraternity, judging from the number of participants in the annual Society tournament. Well over 100 followers of the Scottish pastime answered the call of Chairman Frank Lawrence on Tuesday and played their qualifying rounds. The contestants were grouped in three flights after the qualifying round, the subdivision being based on the qualifying scores for 18 holes. Match play continued in each of these flights until Friday afternoon, when the finals were played. In the Society Championship flight, Claude Foster was opposed by John Warren Watson, the latter winning the match and receiving the medal emblematic of the 1922 Championship. Mr. Watson having won the same honor at West Baden in 1921, it seems proper to suggest that those planning to attend next summer's meeting get plenty of coaching and come prepared to terminate this monopoly. The second-flight prize was taken by Sanford Brown, who battled C. S. Pelton in the finals. D. L. Gallup won the third-flight final match from W. Ray. The driving contest was very close, Jack Gray winning with a drive that was only 2 yd. ahead of Sanford Brown's. C. H. Foster was third. The golf putting contest was won by E. O. Jones, R. A. Watson finishing second and George Case third.

The ladies golf events attracted a much larger field than in years past. Mrs. George Case, who won, found her laurels none too easy to attain. Mrs. J. B. Funk was second in the ladies' golf tournament, with Mrs. C. H. Foster placed third. Interest was added to the ladies putting contest by the opening of a typical pari-mutuel betting establishment where wagers could be placed on one's favorite. As usual, the favorites fell by the boards and Mrs. C. H. Foster won with a comfortable margin from Mrs. George Case and Mrs. J. B. Funk, who were second and third respectively. The clock-golf contest resulted in a triumph for Mrs. Jack Gray.

#### TENNIS TOURNAMENTS CLOSELY CONTESTED

The lovers of tennis who attended the Summer Meeting were rewarded with the opportunity to play on exceptionally fine courts and amidst keen competition. The champion of last year, C. F. Clarkson, met his Waterloo in the person of C. A. Thompson. The two sets of this match were hotly contested and drew a large and appreciative gallery. Mr. Thompson was returned victor, 11-9, 6-4. The doubles match was equally close and interesting, requiring five sets and presenting many exciting rallies. Herbert Chase and Walter Buettner won out over H. M. Crane and C. F. Clarkson, 10-8, 6-3, 1-6, 4-6, 6-3. The ladies tennis championship was won by Mrs. Snead and the mixed doubles were captured by Miss Jessie McKenzie paired with C. A. Thompson.

Lon R. Smith and his enthusiastic clay-bird marksmen enjoyed 4 days of sharpshooting. It was understood that the winner on Friday was to be declared Society Champion and that he would receive the championship medal. W. S. Harley carried off this honor, with W. H. Miller close on the trail. The prize shooters the other three days were W. H. Miller, George Duck and R. M. Owen.

#### SWIMMING EVENTS AROUSE INTEREST

One of the most entertaining features of the sports program, and the one attracting the largest audience, was the swimming meet in the large pool on Wednesday evening. An exceptionally large field of entrants participated in the events and in most instances the races were closely contested. Gordon Brown proved to be very accomplished subaquatically and garnered unto himself the lion's share of the spoils. Many of the stunt races presented amusing competitions, George Briggs particularly surprising the gallery with his brilliant water-golf victory. The ladies came into the spotlight in several races and naturally added color to the program. The complete summary of the swimming races follows:

*Swimming Events*

Inter-Section Relay—First, Metropolitan; second, Detroit
33-Yd. Swim (Men)—First, W. H. Miller; second, W. S. Davidson; third, Neil MacCoull
33-Yd. Swim (Men over 40)—First, Rollin Abell; second, W. S. Harley
33-Yd. Swim (Ladies)—First, Mrs. Ernest Dickey; second, Miss Catherine Cramer
66-Yd. Swim (Men)—First, Gordon Brown; second, W. H. Miller; third, W. S. Davidson
66-Yd. Swim (Ladies)—First, Mrs. J. F. Winchester; second, Miss Anne Koch
Section Balloon Relay—First, Detroit; second, Cleveland
Plunge for Distance—First, T. A. Peck; second, L. C. Hill; third, A. K. Brumbaugh
Egg Race (Men)—First, Neil MacCoull; second, Gordon Brown
Candle Race (Ladies)—First, Mrs. E. Dickey
Blindfold Race (Men)—First, Gordon Brown; second, Mason Rumney; third, F. A. Thompson
Diving Contest (Men)—First, Harold Butcher; second, Neil MacCoull; third, Gordon Brown
Plate Diving Contest (Men)—First, A. C. Bigelow; second, Gordon Brown
Nightshirt Race (Men)—First, Gordon Brown; second, T. A. Peck; third, A. C. Bigelow
Water Golf Race (Men)—First, George Briggs; second, Neil MacCoull; third, F. A. Thompson

#### ANNUAL FIELD DAY A FEATURE

Led by a typical Darktown Alexander's Ragtime Band, the whole assemblage marched to the big field meet on Thursday afternoon and enjoyed the antics of our versa-

tile automotivian athletes. There were jumps, runs, stunts and what-nots to enable the ambitious ones to unkink muscles long since labeled dormant. No casualties were recorded but limps and charley-horses were quite the fashion on Friday. Of course H. E. Kirby pushed the shot many feet beyond his nearest rival, as is his custom; Norma Porter tossed the pellet of national pastime yards beyond any other miss; and Burt Brodt proved that he is still master of the sprints and jumps. The great surprise of the field day was the unheralded speed of Indiana's youthful relay team, which was assembled and groomed by George Briggs. Here are the summaries

*Track and Field Meet*

50-Yd. Dash (Men under 30)—First, B. W. Brodt; second, M. P. Whitney; third, E. O. Jones

50-Yd. Dash (Men 30 to 40)—First, Neil McMillan, Jr.; second, B. S. Pfeiffer; third, W. F. Rockwell

50-Yd. Dash (Men over 40)—First, Mark Smith; second, W. S. Harley; third, L. W. Rosenthal

50-Yd. Dash (Boys under 12)—First, Bobby Germaine; second, Danny Duesenberg; third, Warren Elliot

50-Yd. Dash (Boys under 16)—First, Robert Jardine; second, Gould Klinedinst; third, Danny Duesenberg

50-Yd. Dash (Ladies)—First, Mrs. Ernest Dickey; second, Miss Anne Koch; third, Mrs. Hauser

Fat Man's Race—First, H. L. Williams; second, T. V. Buckwalter; third, G. E. Strohm

Three-Legged Race (Men)—First, Neil McMillan, Jr., and E. O. Jones, second, W. F. Rockwell and B. S. Pfeiffer

Potato Race (Men)—First, E. O. Jones; second, Neil McMillan, Jr.; third, M. L. Hull

Potato Race (Ladies)—First, Mrs. E. Dickey; second, Mrs. Harry Tarantous; third, Miss J. McCormick

One-Legged Race—First, E. P. Warner; second, W. F. Rockwell; third, B. S. Pfeiffer

Three-Legged Race (Mixed)—First Miss Anne Koch and Mr. W. F. Rockwell; second, Miss J. McCormick and Mr. B. S. Pfeiffer

Shot Put—First, H. E. Kirby; second, T. V. Buckwalter; third, I. S. Snead

Standing Broad Jump (Men under 40)—First, B. W. Brodt; second, F. G. Whittington; third, F. F. Kishline

Standing Broad Jump (Men over 40)—First, W. S. Harley; second, J. G. Vincent; third, T. V. Buckwalter

Hop, Skip and Jump—First, B. W. Brodt; second, W. L. Batt; third, W. S. Davidson

High Jump—First, B. W. Brodt; second, M. P. Whitney; third, F. G. Whittington

Throwing Baseball (Ladies)—First, Miss Norma Porter; second, Mrs. Beegle; third, Miss Ruth Porter

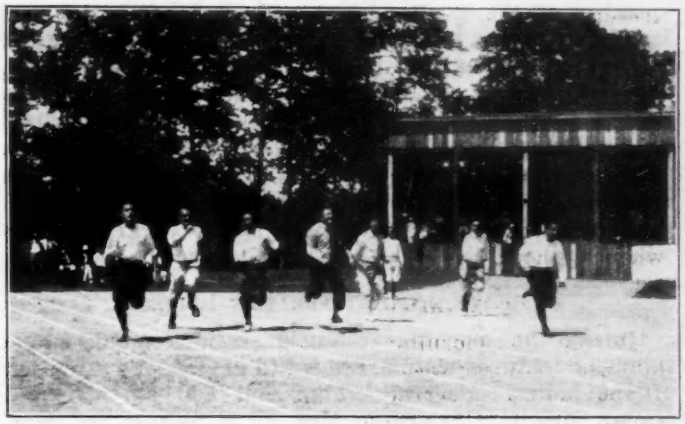
Egg Race (Ladies)—First, Miss J. McCormick; second, Mrs. George Case; third, Mrs. Metz.

Inter-Section Relay Race—First, Indiana, Mark Smith, George Briggs, F. A. Clawsen, M. L. Hull; second, Detroit; third, Metropolitan

The Inter-Section baseball series had many followers and was directly responsible for numerous frozen voices among the ardent rooters. Cleveland defeated Metropolitan on Wednesday 19 to 3, this game being mistaken several times for a fly-chasing contest. Wild heaves, muffs and boots were the order of the day and the least-worst team triumphed. Air-tight ball featured the Detroit-Indiana game on Thursday, the former Section winning 6-3, largely due to the dependable manner in which E. O. Jones garnered flies in left field. Harry Figgie's Cleveland warriors defeated Detroit in the final on Friday, 10-8, the game being very close until the last man was retired. Figgie will hold the Inter-Section Cup in Cleveland until next year, when the riot will be renewed.

### INTER-SECTION CHAMPIONSHIP TO METROPOLITAN

The handsome cup emblematic of the Section Athletic Championship goes to Metropolitan Section for 1922, with a score of 121½ points; Cleveland was second with 118 points; Detroit third with 105 points; and Mid-West fourth with 55 points. The Metropolitan points were collected mostly in the swimming meet and the tennis tournament. Cleveland was proficient in baseball, tennis and golf, while Detroit showed to best advantage in the track events. The plan of encouraging friendly rivalry between the Sections was commended by all of the athletes. It unquestionably adds interest to the entire sports program, results in a larger entry list and builds Section morale. The Inter-Section Championship cup will be contested for again next year and without any doubt the various Section leaders will come well fortified to wrest it from the tribe of Manhattan.

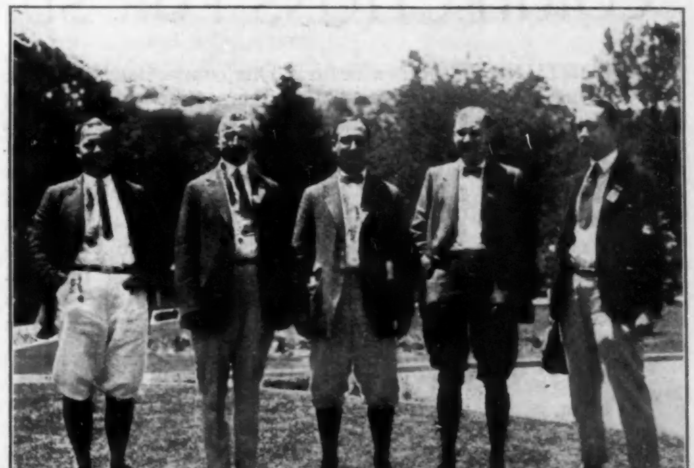


"They're Off"

Another feature of the scoring plan was the awarding of a handsome prize to the individual all-around athletic champion. Gordon Brown, whose swimming was a pleasure to watch, captured this honor largely because of his prowess in the water. His total score was 24½, Burt Brodt being close behind him with 22½ points.

### The Annual Dinner Burlesqued

The Metropolitan Section was responsible for the one big surprise of the meeting. On Thursday evening all of the members dined together and were astonished to find themselves in the midst of a huge burlesque of the 1922 Annual Dinner of the Society in New York City. The printed menus were badly garbled, much liberty being taken with the names of the honored speakers of the evening. Catchy parodies were sung to old familiar tunes and music was provoked by a burlesque song-leader. The



The Meetings Committee

speeches of the evening were patterned after those of last January, but little respect was shown the feelings of those whose names were involved. It was all in good fun, very well worked out and provoked many a laugh. This Section stunt was followed by a wireless concert arranged by the Detroit Section. The demonstration was such a far cry from the real thing that the audience became suspicious of its genuineness in the early stages and absolutely ruined the perpetrator's plans of a grand finale when the hoax was to have been announced.

### Entertainment for the Ladies

During the morning technical sessions many entertainment features were arranged to engage the attention of the ladies. Several bridge and 500 parties were played, Mrs. C. C. Carlton, Mrs. F. S. Slocum and Mrs. J. W. Ruzicka being the winners at bridge on different occasions. The 500 winners were Mrs. Ernest Dickey, Mrs. Jack Gray and Mrs. F. G. Whittington. On Wednesday morning an automobile ride was taken by a large party of the ladies through the surrounding country and the Greenbrier Valley. Of course, there were golf, tennis and other sporting events for the gentler sex; these are described in the section of this report covering the sports. Wireless concerts were provided through the courtesy of the Westinghouse Electric & Mfg. Co. Baritone concerts were given each day by Robert Crawford, who was accompanied by Rockwell Ferris at the piano. This highly complimented feature was arranged by the Cleveland Section. Dancing and motion pictures ran simultaneously in the evenings, the dancing program reaching a climax in the Grand Ball on Friday evening with its interesting formal and jazz dancing contests. The judges had great difficulty in picking the most graceful steppers but made popular decisions in choosing Miss Catherine Cramer as the winner of the formal contest and Mrs. J. B. Funk as the best portrayer of the syncopated gyrations of the present era.

### Gratitude to the Committees

In closing this report of the 1922 Summer Meeting, credit must be given to those who worked arduously, at no little sacrifice of their own pleasure, to make every feature of the program a success. The Meetings Committee, directed by Carl F. Scott, spent many months preparing the details of the technical, sports and entertainment phases of the meeting. Mason P. Rumney selected the members of his Sports Committee many weeks ahead, held several meetings to discuss the athletic events and perfected an organization of committeemen that guaranteed the smooth running of each contest. The commendable result at White Sulphur Springs reflected the value of this forethought. The following members of the Sports Committee deserve full credit for the success of the athletic program:

*Sports Committee*

Mason P. Rumney, *Chairman*
Frank Lawrence—Golf
Lon R. Smith—Trapshooting
C. A. Thompson—Tennis
Neil McMillan, Jr.—Baseball
B. W. Brodt—Track
C. H. Brennan—Swimming
Walter C. Keys—Inter-Section Contest

The entertainment program included several innovations along with the enjoyable dances and card parties of past summers. Wide approval of the entertainment features was evidenced throughout the meeting. Credit for this branch of the work is due to M. C. Horine and Orrel A. Parker. Howard Spohn deserves special mention for his labors as official announcer and for the assistance he rendered the Entertainment Committee.

The *Daily S.A.E.*, that frivolous sheet which in jocose vein kept all informed of the meeting news and scandal, was published this year by the Timken Roller Bearing Co. under the direction of R. E. McKenzie. Those responsible for the 1922 editions of the *Daily S.A.E.* deserve particular mention for their news-gathering ability, the good-natured humor of the personal items and the very attractive appearance of the paper.

The 1922 Summer Meeting was distinctly creditable to all those whose efforts accomplished its success. Few summer gatherings of the Society have excited such general voluntary approval on the part of those in attendance. White Sulphur Springs lacks nothing a first-class meeting place should have. There seemed to be a universal sentiment among the members in favor of returning to this resort for the 1923 meeting.

# CORRECTIONS FOR SUMMER MEETING PAPERS

UNFORTUNATE errors occurred in connection with two of the Semi-Annual Meeting papers that were printed in the June issue of THE JOURNAL. In the paper by Thomas Midgley, Jr., and T. A. Boyd entitled Detonation Characteristics of Some Blended Motor-Fuels, the cuts for Fig. 3 on p. 454 and Fig. 5 on p. 456 were transposed, as would be apparent from the captions. In this same paper on p. 456, an entire line was omitted from the manuscript as submitted by the authors. The following should be inserted between the words "of temperature" in the 11th line of the second column, "the percentage distilled and that of the horizontal axis was in terms of"

The 8th to the 15th lines as corrected should read as follows:

> axis. From this curve the percentages of the fuel distilling in each 10-deg. interval were obtained, and these values were plotted on a chart in which the scale of the vertical axis was in terms of the percentage distilled and that of the horizontal axis was in terms of temperature. The average boiling-point of the fuel was taken as the point at which a perpendicular passed through the center of gravity of the area enclosed under this differential distillation-curve cut the horizontal or temperature axis.

The caption for Fig. 2 of the paper by H. M. Crane entitled New System of Spring-Suspension for Automotive Vehicles on p. 464 should have indicated that the spring-suspension was applied to a passenger-car chassis.

# The Sections Lunch at the Summer Meeting

THE Sections Luncheon at White Sulphur Springs attracted a very representative group of members interested in the administration of the 11 Sections of the Society. Every Section was represented by at least one of its members and all of the Sections Committee were present. Each of the Section representatives was asked to present a brief summary of his Section's activities during the past year, to outline its plans for the coming fall and winter, and to discuss any problems whose solution seemed essential to the future success of his Section. Section dues, affiliation with local engineering bodies, the relative attractions of technical and non-technical papers and the important matter of securing increased Section membership and larger attendance at meetings, were the major topics of discussion.

A. K. Brumbaugh, chairman of the Sections Committee, presided at the luncheon. He asked the Section officers to place their problems before the meeting in order that there might be a free interchange of experience and that the Sections Committee might be in a position to make recommendations to the Council on any questions that deserved such consideration. Hugh R. Corse spoke briefly for the Buffalo Section. Orrel A. Parker, chairman Cleveland Section, discussed the matter of Section dues and felt that the plan of making all Society members Section members without additional dues, had many good points to recommend it. J. H. Hunt, chairman Dayton Section, said that his Section was fortunate in having the facilities of the Dayton Engineers Club at its disposal. Dayton meetings were well attended, good engineering papers always available and no difficulty experienced in securing Section members. Mr. Hunt believed in the preparation of programs and papers well in advance and stated that all the new committees of the Dayton Section were organized and functioning at the present time for the coming year.

The Detroit Section has just completed a very successful year according to George E. Goddard, its new chairman. Mr. Goddard outlined the program that had attracted such satisfactory attendance and mentioned particularly the great interest shown in production engineering subjects. He recommended strongly the scheduling of meetings by the other Sections on machine tools, manufacturing methods and means of attaining decreased production costs. K. K. Hoagg, vice-chairman, Detroit Section, represented his Section recently in a series of conferences that led to an allied organization of engineering society sections in Detroit. Mr. Hoagg said the Detroit Section of the Society did not affiliate with the other organizations because the cooperative nature of the joint meetings called for the presentation of papers on general rather than specialized subjects. The Detroit Section felt it could not conduct such meetings so that they would be of real constructive value to Society members and that there was danger also of losing the Section's identity and scattering the interest of the Section members.

There appears to be some sentiment in the Indiana Section for the waiving of Section dues according to its chairman, O. C. Berry. Mr. Berry was strongly opposed to the use of Section funds for the provision of professional entertainment at Section meetings. He felt that money thus spent could be used to much better advantage in securing able speakers and providing attractive engineering programs. The Indiana Section intends to organize its meetings well in advance and use them as an inducement in the solicitation of new Section members.

Taliaferro Milton, chairman, Midwest Section, discussed the plans of his Section to overcome the handicap of diversified employment and widely scattered geographical location of the members in Chicago and its environs. The Section has organized a large and active meetings committee. They are hoping to secure men of national reputation in the automotive field as speakers at their meetings. The topics are to be of general rather than specific technical interest. Highly technical meetings have not attracted satisfactory numbers and are believed unsuitable in the Chicago district because of the aforementioned diversification of automotive interest in this territory. Mr. Milton expressed a favorable opinion of section membership without added dues.

The Minneapolis Section finds a predominating interest in tractors and power-farming in its district, according to L. A. Emerson, who represented the Section at the luncheon. Naturally, interest in the Minneapolis meetings had suffered somewhat from the recent depression in the agricultural implement business, but the new officers are finding that the return of prosperity has awakened new and greater enthusiasm in their district and are confident of having an active year for the Section. During the luncheon, the following telegram was received from Phil Overman, secretary, Minneapolis Section, as evidence that things in the northwest are not quiescent.

GREETINGS FROM MINNEAPOLIS SECTION

> Our Summer Meeting and picnic today successful demonstration advantage social contact. Repetition as annual event assured.

R. E. Plimpton, secretary, Metropolitan Section, summarized the program of meetings held in New York City during the past year and mentioned the great interest shown in the subject of motor rail-cars. He said the plans for the series of meetings for the coming season were completed and speakers selected. The Metropolitan Section has organized a joint membership and reception committee the function of which is to introduce non-Section members at the meetings soliciting their Section membership at the same time. This suggestion is worthy of adoption by all of the Sections. It should be more productive of results than appeals by letter since the average prospect appreciates the personal interest shown in him.

It has been the experience of the New England Section officers that highly technical papers do not attract as large an attendance as those on general and non-technical subjects. R. E. Northway, who represented the Section, recommended that the Sections turn their attention to subjects not treated comprehensively in the national meetings, the manufacture of tires being an example. Sentiment in New England favored the waiving of Section dues. Mr. Northway did not believe Section membership could be built up satisfactorily except by personal solicitation.

The Washington Section was represented by Conrad H. Young, its treasurer. He said that attendance at their meetings was necessarily limited by the comparatively small Society membership in the district. The papers at Washington meetings are always of excellent calibre, many of them being presented by scientists and engineers in the Government service. However, it is difficult to secure the release of these papers for publication. The problem of collecting Section dues confronted the Washington officers but they were not convinced that waiving them would be an advisable step.

T. F. Cullen, chairman, Pennsylvania Section, discussing the matter of Section dues felt that the members showed more interest in the Section when they had to pay for the privilege of membership. The possibility of noon meetings had been proposed in the Pennsylvania Section but was dismissed in favor of the customary evening meetings after close study. Mr. Cullen favored interesting local automotive associations in the Section meetings and inviting their members as guests to increase the attendance, later soliciting for

membership in the Society any of those who are qualified.

Past-President David Beecroft recommended that greater thought be given by the officers to the staging of meetings. He did not believe our members appreciated or favored professional entertainment. The work of administering the Section's affairs should be well distributed, not shouldered by two or three officers. Mr. Beecroft favored having prominent local business men as guests at Section dinners. The Section should be a factor in all things automotive in its district and the business men should recognize its position in this respect. Service should be discussed in a local way, also automotive legislation. The Sections should each maintain close contact with the universities and educational institutions in their locality. Large meetings should be held in conjunction with the local automotive shows. It was Mr. Beecroft's belief that Section dues were not an obstacle if the meetings were made sufficiently attractive to the members.

President Bachman, though of an open mind on the matter of Section dues, did not believe the waiving of them would increase interest in the meetings. He cited cases he had personally investigated to substantiate his belief. He favored affiliation with local engineering bodies but only when the Society's identity was not submerged. Mr. Bachman hoped that the new Sections officers would recognize as one of their most important duties, the conduct of the election of delegates to the Nominating Committee of 1923. This committee is charged with what is virtually the selection of general Society officers and the election of its members should be conducted in the most serious and earnest manner.

A number of valuable suggestions were made in the roundtable discussion following the short talks. Hugh Corse believed it advisable that the membership committee of each Section include a man of sales instinct able to formulate convincing letters for membership and dues solicitation. R. J. Nightingale likened the Sections to the roots of the national tree and emphasized the importance of keeping them healthy. He favored production meetings, and cooperation with other local engineering groups. Thos. J. Lítle, Jr., suggested a closer contact with the automotive companies in each Section territory, posting notices of the meetings in their factories and asking that they request their executives to attend the Section meetings. Conrad H. Young believed the dealer in automotive products could be interested more strongly in the Section meetings. George Goddard made the suggestion that the Sections especially invite to their dinners those pioneer members of the Society who by reason of attaining positions of great responsibility found it difficult to attend Section meetings.

A. K. Brumbaugh thanked the several Section officers for their suggestions and constructive thought, on behalf of the Sections Committee. He impressed them with the necessity of recognizing the responsibility that was placed upon them by the members. Without such recognition and an accompanying sacrifice of personal time, there could be anticipated only a mediocre success of the Sections. The Sections Committee is anxious to have every problem faced by the Sections placed in its hands for careful consideration and is certain that with the cooperation of the Council and Section officers there is no obstacle that cannot be surmounted.

Those who attended the luncheon were:

| | |
|---|---|
| B. B. Bachman | J. H. Hunt |
| David Beecroft | Thos. J. Lítle, Jr. |
| O. C. Berry | H. O. K. Meister |
| A. K. Brumbaugh | Taliaferro Milton |
| Coker F. Clarkson | R. J. Nightingale |
| Hugh R. Corse | R. E. Northway |
| G. W. Cravens | Orrel A. Parker |
| T. F. Cullen | B. S. Pfeiffer |
| L. A. Emerson | R. E. Plimpton |
| G. W. Gilmer, Jr. | H. L. Pope |
| George E. Goddard | H. W. Slauson |
| L. Clayton Hill | E. W. Weaver |
| K. K. Hoagg | Conrad H. Young |

# CURRENT STANDARDIZATION WORK

(*Concluded from page* 118)

Batteries, p. B23 of the S. A. E. HANDBOOK, and are dimensions for the type of unit container that represents a new development in jar construction. The specifications follow.

(1) *Height.* Containers shall be made in two heights only; the B height for plates approximately 4¾ in. high and the C height for plates approximately 5¼ in. high.

(2) *Plate Supporting Ribs.* There shall be four ribs in each compartment on 1⅜-in. centers. The rib height shall be ¾ in. for B-height containers and ⅞ in. for C-height containers.

(3) *Top of Ribs to Top of Container.* These heights shall be

B-height containers..............6½ in.
C-height containers..............7 in.

(4) *Inside Width of Compartments.* The inside width of the compartment shall be 5 31/32 in. with tolerances of plus or minus 1/32 in.

(5) *Inside Lengths of Compartments.*

(a) Six-Compartment Containers

| *B Height* | | *C Height* | |
|---|---|---|---|
| S-3-B | 1 5/16 in. | S-4-C | 1½ in. |
| S-4-B | 1½ in. | S-5-C | 1 11/16 in. |
| S-5-B | 1 11/16 in. | | |

(b) Three-Compartment Containers

| *B Height* | | *C Height* | |
|---|---|---|---|
| S-7-B | 2 1/16 in. | S-8-C | 2⅜ in. |
| S-8-B | 2⅜ in. | S-10-C | 2 13/16 in. |
| S-9-B | 2 7/16 in. | S-13-C | 3¼ in. |
| S-10-B | 2 13/16 in. | S-14-C | 3 5/16 in. |
| S-16-B | 3 11/16 in. | S-16-C | 3 11/16 in. |
| S-18-B | 3 15/16 in. | | |
| S-19-B | 4⅛ in. | | |

(6) *Partitions Between Compartments.* The thickness of partitions between compartments shall be 3/16 in. minimum and ¼ in. maximum.

There was considerable discussion as to what the unit-battery container should be called. The following suggestions were submitted: cellbox, jarbox, cellblock, unit-container, integral container, container-in-block and monoblock container. The last name was finally decided upon as being the most descriptive and in accord with the use of the same term in cylinder construction.

It is not considered advisable to specify the dimensions for the outside of the container, since this would have a tendency to limit future developments. It was thought, however, that the dimensions agreed upon would serve as a very good basis for determining the ultimate outside dimensions.

The understanding was that the dimensions determined upon would be submitted to battery and hard-rubber manufacturers for further comment before final Division action is taken.

### WIRE MESH

At the May 9 meeting of the Parts and Fittings Division it was stated that the wire-mesh manufacturers would like to have a standard established because at the present time each manufacturer makes practically any mesh that is ordered, one company alone making many hundreds of meshes of different wire sizes and mesh.

# Applicants *for* Membership

The applications for membership received between May 18 and June 15, 1922, are given below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ADAMS, JOHN NEWELL, student, University of Michigan, *Ann Arbor, Mich.*

ANTHONY, JOHN EDWARD, designing engineer, International Harvester Co., *Chicago.*

AUSTIN, E. W., district manager, Timken Roller Bearing Co., *Cleveland.*

BARTLETT, KING S., automobile mechanic, 770 South Commercial Street, *Salem, Ore.*

BARTHOLOMEW, J. R., assistant engineer, Westinghouse Pacific Coast Brake Co., *Emeryville, Cal.*

BASCH, JACOB JUSTIN, sales engineer, G. & O. Mfg. Co., *New Haven, Conn.*

BAUCH, CHARLES H., Navy Department, *City of Washington.*

BITTERMAN, SIMON, president and manager, American Auto Products Co., *Denver, Col.*

BREWER, HENRY, district sales manager, Leeds & Northrup Co., *Philadelphia.*

BUCKBEE, GEORGE A., salesman, Gurney Ball Bearing Co., *Jamestown, N. Y.*

BURT, LEO O., designer, Chevrolet Motor Co., *Detroit.*

BUTTERICK, W. B., chief mechanic and superintendent, Miller North Broad Storage Co., *Philadelphia.*

CHAMPION, E. M., works manager, American Motor Body Co., *Philadelphia.*

CLARK, WILLARD T., resident partner, Henry W. Peabody & Co., *Buenos Aires, Argentine Republic.*

CRAWFORD, KENNETH G., draftsman, Sanford Motor Truck Co., *Syracuse, N. Y.*

CROSS, CHARLES H., student, Ohio State University, *Columbus, Ohio.*

DANLY, PHILO H., engineer, tractor works, International Harvester Co., *Chicago.*

DOYLE, WILLIAM EDWARD, JR., student, Stevens Institute of Technology, *Hoboken, N. J.*

DUFFEE, FLOYD W., assistant professor, University of Wisconsin, *Madison, Wis.*

ERSKINE, ALBERT R., president, Studebaker Corporation of America, *South Bend, Ind.*

FOSTER, WILLIAM J., aeronautical mechanical engineer, Air Service, McCook Field, *Dayton, Ohio.*

FRENCH, C. A., engineer, International Harvester Co., *Chicago.*

GABBER, JACOB E., student, Ohio State University, *Columbus, Ohio.*

GARY, C. E., draftsman, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

GINGHER, D. B., automobile mechanic, H. B. Tait & Co., *Columbus, Ohio.*

GUNN, GEORGE, JR., president and manager, Kelly-Springfield Truck Sales Co., *Seattle, Wash.*

HAMPTON, E. H., sales engineer, Hampton-Watson & Cia, *Buenos Aires, Argentine Republic.*

HERMANN, JOHN F., assistant engineer, Phelps Light & Power Co., *Rock Island, Ill.*

HINKLEY, RAY A., engine designer, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

HOLMES, JOHN Q., metallurgical engineer, Nordyke & Marmon Co., *Indianapolis.*

HORTHY, WILLIAM A., mechanical engineer, tractor works, International Harvester Co., *Chicago.*

HOUSE, BRYAN E., designer, Maxwell Motor Co., *Detroit.*

JACKSON, E. F., manager manufacturers' sales, Goodyear Tire & Rubber Co., Inc., *Detroit.*

KELLEY, GEORGE L., metallurgist, Edward G. Budd Mfg. Co., *Philadelphia.*

KENNEDY, JAMES T., sales department, Goodyear Tire & Rubber Co., *Detroit.*

KNOWLES, FRANK LESTER, student, Ohio State University, *Columbus, Ohio.*

KNOWLTON, H. B., instructor, Central Continuation School, *Milwaukee.*

KOLB, GEORGE F., manager, motorcycle division, Bullard Machine Tool Co., *Bridgeport, Conn.*

LADAIR, THOMAS F., automobile distributor, 504 Cass Street, *Milwaukee.*

LAWRENCE, F. W., salesman, A. O. Smith Corporation, *Detroit.*

LEIGHTON, LIEUT. BRUCE G., bureau of aeronautics, Navy Department, *City of Washington.*

LEWIS, MAJOR BURTON O., Ordnance Department, *City of Washington.*

LONG, RAY A., chief engineer, Columbia Motors Co., *Detroit.*

LUTZENBERGER, LOUIS DICKSON, student, Ohio State University, *Columbus, Ohio.*

MCCORMACK, MERLE H., student, University of Michigan, *Ann Arbor, Mich.*

MCFAWN, FRED, district sales engineer, Stanley Works, *New Britain, Conn.*

MCWHIR, DAVID, chief inspector, Wright Aeronautical Corporation, *Paterson, N. J.*

MORTIMER, BRUCE G., assistant superintendent, garage service station, Wilson Motor Sales Co., *Toronto, Ont., Canada.*

MRAZ, EMIL, superintendent, Temme Spring Corporation, *Chicago.*

NATIONAL MALLEABLE CASTINGS CO., 10600 Quincy Avenue, *Cleveland.* (Affiliate member.)

PALMER, H. A., chief engineer, Reynolds Spring Co., *Jackson, Mich.*

PLEISS, PAUL, secretary and director, Budd Wheel Co., *Philadelphia.*

PLUMRIDGE, TOM G., consulting engineer, American Technical Society and American School of Correspondence, *Chicago.*

PRODOEHL, H. G., charge of tool design department, International Harvester Co., *Chicago.*

RAHUSEN, E. N., manager, Handelsbureau voor Automobiel & Vliegtuig-Industrie, *Ryswyk, Holland.*

RAGSDALE, R. J. W., engineer, Budd Wheel Co., *Philadelphia.*

REESE, EDWIN KENNETH, sales manager, vice-president, King Tool Co.; salesman and production manager, Packard Engineering Co., *Cleveland.*

RICARDO, HARRY R., consulting engineer, Ricardo & Co., Ltd., *Old Shoreham, Sussex, England.*

RICHARDSON, ARCHIBALD P., sales manager, Gurney Ball Bearing Co., *Jamestown, N. Y.*

ROUSE, GEORGE ALAN, assistant superintendent motor vehicles, Standard Oil Co. (N. J.), *Baltimore.*

SAVANT, A. K., student, University of Michigan, *Ann Arbor, Mich.*

SCULLY, JAMES N., president, Houdaille Co., *Buffalo.*

SEILER, PAUL W., president and general manager, Ternstedt Mfg. Co., *Detroit.*

SKINNER CO., LTD., *Gananoque, Ont., Canada.* (Affiliate Member)

SKURRAY, ERNEST CLEMENT, motor engineer, Skurray's, *Swindon, Wiltshire, England.*

SMITH, LEOPOLD J., draftsman, American Railway Express Co., *New York City.*

SMITH, STANFORD ALLEN, chief inspector, Lexington Motor Co., *Connersville, Ind.*

SMITH, T. D., sales manager, Harvey Electric Co., *Chicago.*

STINSON, KARL W., instructor in mechanical engineering, Ohio State University, *Columbus, Ohio.*

STOVER SIGNAL ENGINEERING CO., *Racine, Wis.* (Affiliate Member)

SUN CO., Finance Building, *Philadelphia.* (Affiliate Member)

SUTHERLAND, J. D., general sales manager, Wyman-Gordon Co., *Worcester, Mass.*, and *Harvey, Ill.*

TALL, G. W., JR., sales manager electric furnace division, Leeds & Northrup Co., *Philadelphia.*

TICHY, V. L., assistant metallurgist, White Motor Co., *Cleveland.*

TIMKEN, H. H., president, Timken Roller Bearing Co., *Canton, Ohio.*

VOORHEIS, GLENN IRVING, student, Michigan Agricultural College, *East Lansing, Mich.*

WAGNER, LEONARD J., draftsman, Ohio Body & Blower Co., *Cleveland.*

WELCH, STANLEY P., New York branch manager, Kelsey Motor Co., *Newark, N. J.*

WHARTON, THOMAS P., president, Wharton Motors Co., *Johnstown, Pa.*

WILSON, CHRISTIAN, president and chief engineer, C. Wilson Co., *Cliftondale, Mass.*

WILLS, C. HAROLD, president, C. H. Wills & Co., *Marysville, Mich.*

ZIMMERMAN, M. A., student, Ohio State University, *Columbus, Ohio.*

ZIMMERMAN, PAUL G., engineer, Aeromarine Plane & Motor Co., *Keyport, N. J.*